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APRIL ★ 1942

INDUSTRIAL
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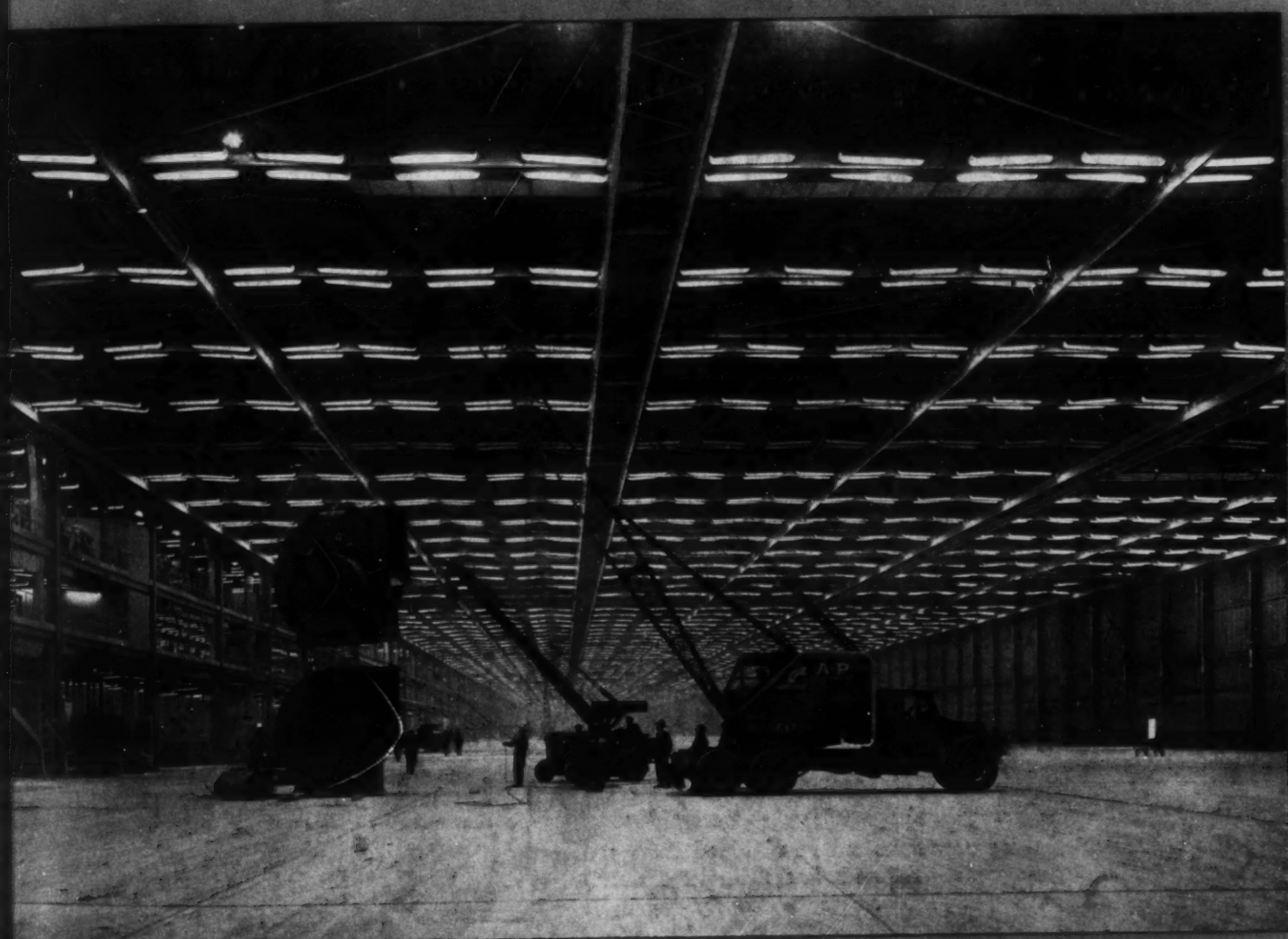
Page 243—Journal Section, American Society of Heating and Ventilating Engineers



Two large fans go into position high above the main assembly aisle in a new air conditioned war production plant

CONTENTS..PAGE 35

HERE'S WHY

TEXROPE* SUPER-7 V-BELTS ARE 4 WAYS BETTER!

1 **50% STRONGER CORDS!** Sensational new Flexon process makes cords 50% stronger than even those in former great Texrope belts.

2 **20% MORE CORDS!** Count them! You get 20% more of these tough, extra pulling power cords in every Texrope Super-7.

3 **INCREASED LIFE!** Cords float on amazing cool-running cushion rubber that absorbs shock . . . actually increases belt life phenomenally.

4 **EXTRA PROTECTION!** Exclusive duplex-sealed cover protects pulling cords against dirt, grit, moisture, and other adverse elements.

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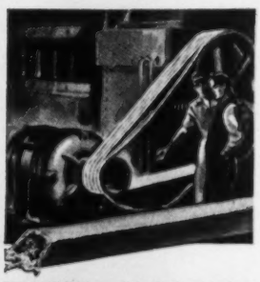
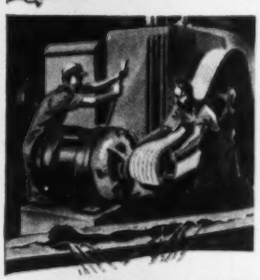


LOOK OUT FOR DANGER in belts with stiff, unpliable cords! Such belts *look* strong . . . but actually they buckle over sheaves, build up excessive heat that attacks pulling cords. Result: Belt failure far sooner than you expect it.



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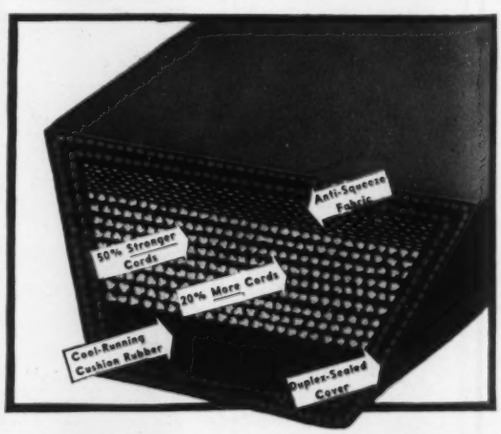

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IN THIS ISSUE

Volume 14 April, 1942 Number 4

The Editor's Pages (news notes and comment).....front section 31

Code for Pressure Piping (District Heating Piping Systems, by G. K. Saurwein; Refrigeration Piping Systems, by A. B. Stickney)..... 215

Flow Meters Reveal Waste of Steam (brief note), by S. D. Distelhorst.. 219

The Man Behind the Man Behind the Man Behind the Gun—the Air Conditioning Engineer (how a big new plant is air conditioned), by F. O. Jordan..... 220

Floor Coils Heat Auditorium of New Community Church (radiant heating used in unusual building)..... 223

Extension of Ratings by Contractors Facilitated (priorities news item).. 225

Air Conditioning—for Offense—Assures Uninterrupted Production of War Material (picture story on huge air conditioned plant)..... 226

"Open for Discussion"—communications to the editor on protecting steel heating and cooling coils, and on high velocity air circulation in armories 228

War Problems Forum (short articles and items of particular value to those concerned with heating, piping and air conditioning for production for victory) 229

How Piping Is Welded for Navy Service (welding plays important part aboard the ships of our fleet)..... 235

Change in Piping Corrects Trouble with Boiler Water Level (solution of a practical piping problem briefly explained), by R. S. Hawley..... 236

Air in Contact with Water (analyzing air conditioning and other heat exchange problems with the psychrometric chart), by William Goodman 237

What's Wrong with This Heating System? (Samuel R. Lewis answers a reader's question) 240

Graphical Analysis of Periodic Loads (HPAC data sheet presents information not found in the textbooks), by B. F. Raber and F. W. Hutchinson 241

Equipment Developments (new products you'll want to know about).....back section 107

Trade Literature (helpful catalogs and bulletins).....back section 117

Useful Information (new books, booklets, reports, etc.).....back section 122

Coming Events (note them on your calendar).....back section 126

Index to Advertisers.....back section 128

Journal Section of the American Society of Heating and Ventilating Engineers

The Effect of Attic Fan Operation on the Cooling of a Structure, by W. A. Hinton and A. F. Poor..... 244

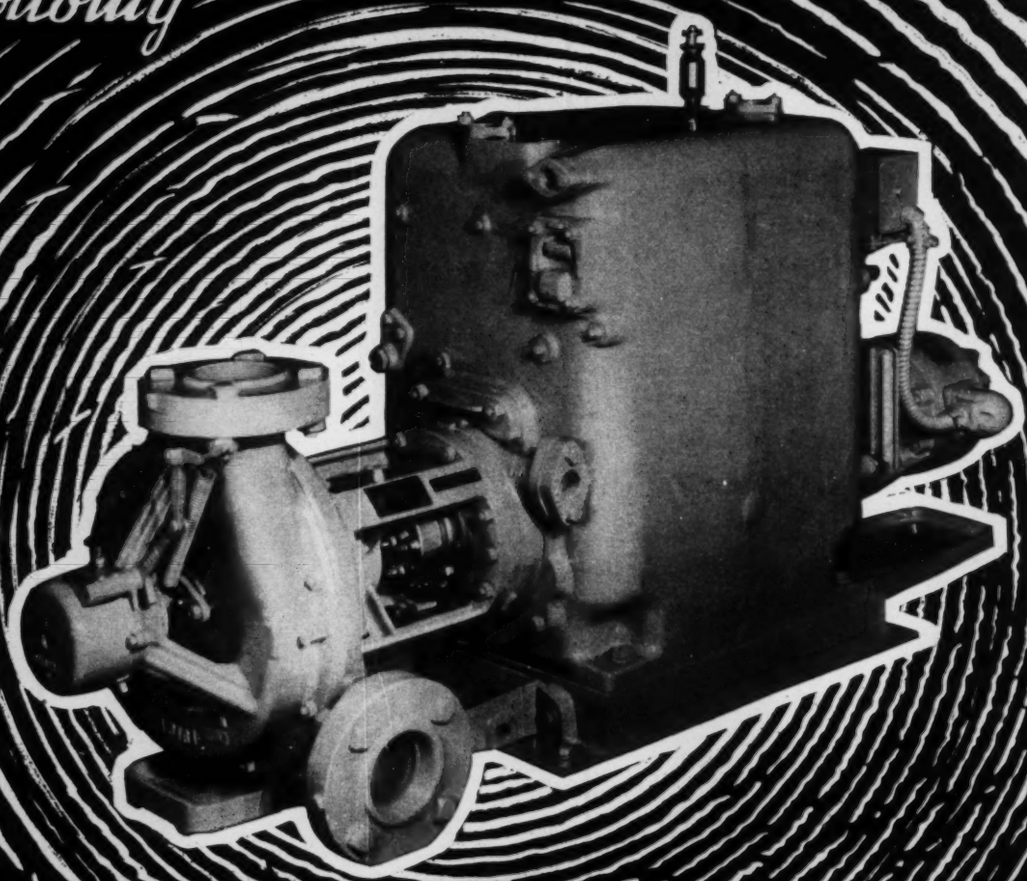
The ASHVE in the War, by S. H. Downs..... 251

Observed Performance of Some Experimental Chimneys, by Richard S. Dill, Paul R. Achenbach and Jesse T. Duck..... 252

Proceedings of Annual Meeting 1942..... 262

Bulletin Section 272

Centralized Heating Economy



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CODE FOR PRESSURE PIPING

DISTRICT HEATING PIPING SYSTEMS

By G. K. Saurwein, Chairman, Subcommittee on District Heating Piping; Superintendent, Engineering Department, Harvard University

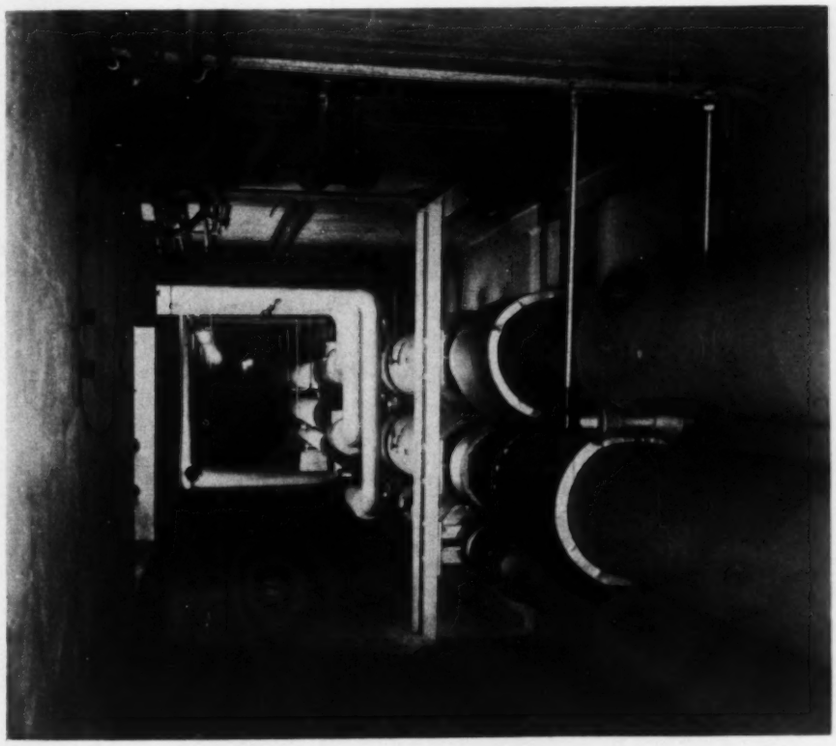
SUMMARY—Since the ASA code for pressure piping was first issued as a tentative standard in 1935, there have been many developments and advances in piping practices, as described by Sabin Crocker in the January HPAC in the first of a group of articles, of which this is the fourth. The piping code has been thoroughly revised and is being reissued as a full American standard. Section 1 of the new code, on power piping, was reviewed by Alfred Iddles in the February HPAC. Last month, the section on gas and air piping was covered by John S. Haug, and the section on oil piping systems was analyzed by A. D. Sanderson. . . . This month, the code sections on district heating piping, and on refrigeration piping (including air conditioning) are reviewed by the respective subcommittee chairmen

THE DISTRICT heating section of the American Standards Association code for pressure piping (B31) follows closely the requirements of section 1 on power piping, since both sections cover piping intended to convey steam or hot water. While pressures used in district heating piping have not attained and are not likely to reach the higher pressures and temperatures currently employed in steam power plants, there has been some increase in service conditions for district heating. In this revision of section 4, therefore, the scope has been expanded to include pressures not exceeding 600 psi and temperatures not in excess of 750 F instead of the 400 psi and 750 F limits previously specified.

In some recent installations where steam has been conveyed from utility power plants to nearby manufacturing plants, the new service limitations have been exceeded. While these are more or less isolated cases of special industrial connections and cannot be considered as "street steam" lines, it is the intent that this code provide for such lines. Accordingly, it is provided that piping falling within the classifications set up in this section, and carrying steam in excess of 600 psi and 750 F temperature, must be designed in accordance with the requirements of section 1, on power piping, of the code.

A further clarification in the scope concerns outdoor piping for industrial plants which is now covered by the following paragraph intended to apply to piping between buildings forming an integrated establishment:

Outdoor steam or hot water distribution piping on the premises of an industrial establishment shall conform to the requirements of this section. Such distribution piping constructed within the buildings of such establishment shall conform to section 1 of this code.



Expansion joints and anchoring stations of the district heating piping system at Harvard university

Valves

In the previous issue of the code, it was specified that gate valves only should be used as stop valves in steam mains. Gate valves were specified because they provide an unobstructed opening in the line without impeding the flow of condensate. Globe valves, by reason of their inherent design as a throttling type of valve, do not permit free flow of condensate. Further, when the globe valve is opened, slugs of water may be projected forward at high velocity with the steam. The resulting shock to the line may cause serious damage.

Plug and Angle Valves—Manufacturers of plug type valves objected to the restriction that no valve other than a gate could be used as a stop valve in steam mains. It was contended that other types, such as plug valves, theoretically could meet the requirement that the flow of condensate should not be

impeded. Since plug valves have been used to a limited extent in district heating piping with satisfactory results, it was agreed to revise the requirement concerning gate valves as follows:

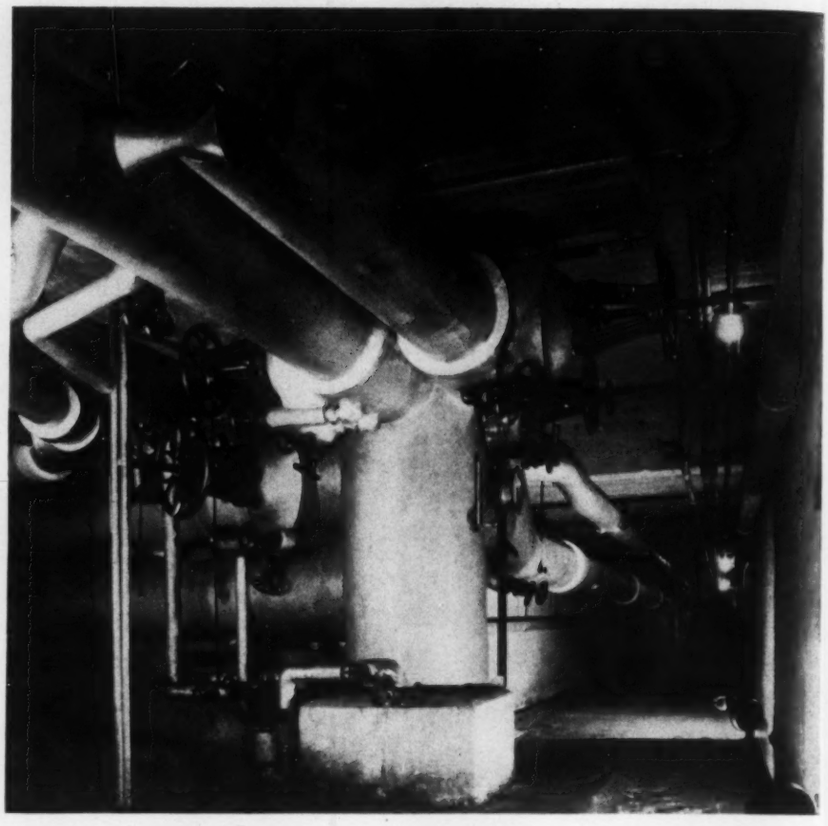
Stop valves used in steam mains shall be constructed and installed so that the drainage of condensate along the bottom of the pipe is not obstructed.

In addition to permitting use of plug valves for conditions where they are suitable, this wording also is understood to include the use of angle valves which, in some locations, may be quite acceptable for use as stop valves.

Reducing and Relief Valves—Reducing and relief valves constitute an important item in district heating piping since it is customary practice to distribute steam at high pressure, and reduce the pressure in the consumer's premises for use in heating equipment which is designed, in general, for low pressure service. In recent years, however, some industrial building heating equipment has been designed to withstand steam pressures up to 100 psi or above, so that it is no longer practical to set a low pressure limit which shall not be exceeded in building heating systems. The previous issue of this section assumed that a steam pressure of 50 psi could not safely be exceeded in heating systems. In view of recent developments, requirements regarding pressure reducing and relief valves in consumer's premises have been revised as follows:

a) Where the street pressure exceeds the safe working pressure of the building heating apparatus, a pressure regulating valve shall be provided near the point of supply to regulate the pressure on building heating equipment within safe limits. In the case of cast iron radiation this pressure shall not exceed 50 psi.

Where the street pressure exceeds 50 psi and is above the safe working pressure of the building steam-using apparatus, a relief valve or valves set at the safe working pressure of the building steam apparatus shall be provided, except that where two pressure reducing valves are installed in series, both set at or below the safe working pressure of equipment served, no relief valve is required. If the installation of a relief valve is not feasible, a trip stop valve set to close at the maximum safe working pressure shall be installed. When the building is continuously attended, an alarm valve or signal may be installed in lieu of a relief or trip stop valve.



Twenty-four inch vertical header, Harvard university system. This piping is within the scope of the district heating piping section of the code for pressure piping

b) The capacity of relief valves shall be such that the pressure rating of the lower pressure piping and equipment shall not be exceeded with full steam flow from the relief valves. Relief valves shall be vented to the atmosphere, and proper protection shall be provided to prevent injury or damage caused by escaping steam.

Particular attention is directed to the requirement that where two pressure reducing valves are installed in series, both set at or below the safe working pressure of equipment served, no relief valve is required. In view of the difficulty of installing relief valve vent risers in the skyscraper type of building, it was considered that the double reducing valves would provide an acceptably safe installation. The requirement on capacity of relief valves has been revised from the previous provision which permitted 33⅓ per cent overpressure. If the latter exceeds the safe working pressure of the system, it is obvious that a hazardous condition would occur when the relief valve operated.

Drains, Drips and Steam Traps

Both the power piping and the district heating sections have included the following provision: "Where two or more drips operating at the same pressure discharge into the same trap, a check valve and a stop valve shall be placed in each line." After further consideration of this requirement by the committee, it was felt inadvisable tacitly to sanction discharge of two or more drips into the same trap, even though a check valve and stop valve are placed in each line.

There is a definite possibility that steam may reach the trap through one drip line faster than through the others, which—in extreme cases—could hold back condensate in the latter. This is commonly referred to as short circuiting, and the solution of the problem is to use a separate trap on each line, which is preferred practice in any case. It was agreed, therefore, to delete the requirement quoted above in both the power and district heating sections, thereby removing any suggestion of a practice which, under some conditions, could be definitely hazardous.

Hydrostatic Tests

While in most instances the district heating section requirements are identical with the corresponding provisions of the power section, one significant point of difference concerns the test of piping after installation. In the case of any underground district heating piping which is to be buried or otherwise made inaccessible, the assembled lines must be tested hydrostatically at a pressure of *one and one-half* times the maximum allowable service pressure. Welded steam piping covered by the power piping section must be tested hydrostatically to at least *two times* the normal service pressure.

In any case, the hydrostatic test is an inadequate means for determining whether a steam line will operate satisfactorily under the desired service conditions. In the instance of district heating piping, it was felt that a hydrostatic test of one and one-half times the service pressure was commensurate with the temperature conditions encountered, and with the hazards involved in underground piping.

General Requirements

Other changes have been made throughout this section to conform to the revisions which were described in the January installment on changes common to all sections of the code, and in the February article concerning the power section of the code. Numerous editorial revisions have been made to bring the section up to date as regards new ASTM specifications and ASA standards which have been developed. Specific requirements follow the classifications as set up in the previous issue of the pressure piping code, except that new requirements have been included for pressures up to 600 psi instead of the limit of 400 psi in the 1935 issue.

REFRIGERATION PIPING SYSTEMS

By A. B. Stickney, Chairman, Subcommittee on Refrigeration Piping Systems; Engineer, Armour & Co.

SECTION 5 of the code covers refrigerant and brine piping, wheresoever used and whether erected on the premises or factory assembled. It does not cover shell apparatus, compressors or pumps, but does include all connecting refrigerant and brine piping, starting at the first joint adjacent to such apparatus. It does not cover service water piping, for which reference is made to the service water requirements of section 1 on power piping.

Although the refrigeration section is new to this code, piping for refrigerants has been extensively covered in the safety code for mechanical refrigeration, ASA B9-39. In order to avoid conflict between the two codes, the B9 provisions applying to piping, which represent known safe practice, have been incorporated in this section as nearly verbatim as possible, and the corresponding B9 paragraph number is given at the end of each such incorporated paragraph as a means of identifying the source of the material. While this section conforms to the general requirements common to other sections of the code previously described, such as reference to latest editions of standards and specifications, inclusion of the same pipe wall thickness formula, etc., there are certain fundamental requirements which are pertinent to this section only and differ from the provisions contained in other sections.

Specific recognition is accorded in this section, for instance, to "manufacturers' standards of long standing" which are permitted for the particular refrigerant service listed by the manufacturer. While such standards must be at least of equal strength and tightness, and capable of withstanding the same hydrostatic test requirements as the American standard for the pressure and temperature, specific reference to MSS (Manufacturers Standardization Society of the Valve and Fittings Industry) or American standards for the service has been impractical because none exist for many of the services.

Flange facings, however, are restricted in any case to the retained gasket type such as the male and female, tongue and groove, or ring joints.

Low Temperatures

Another fundamental point of difference of the section (as distinguished from other sections of the code) concerns the use of pipe, valves, fittings, etc., at the low temperatures encountered in refrigeration practice. Primary ratings for carbon steel flanges, valves, and fittings with ordinary facings used for steam or oil service are designated at 750 F. As the temperature for these services decreases below 750 F, progressively increasing ratings are permitted down to 100 F by the American standards in view of the lesser hazard involved, and as a result of the increase in yield point as the temperature decreases.

For temperatures below zero degrees Fahrenheit, which are common in refrigerating piping systems, the material may become seriously brittle, and special provision must be made to recognize this fact. Accordingly, the following allowances for brittleness of materials are made for temperatures below 0 F:

Cast iron, wrought iron and carbon steel, except as they may qualify under the rule for *other materials,* shall have the design pressure increased 2 per cent for each degree below 0 F, and shall not be used below —60 F.

Copper, brass and bronze, no adjustment.

Other materials must have, at the lowest working temperature, a Charpy impact value of not less than 15 ft-lb on a specimen 10 mm wide, or 10 ft-lb on a specimen 5 mm wide, or 5 ft-lb on a specimen 2.5 mm wide, as determined according to ASTM E-23, using a keyhole notch. As to materials, a footnote says:

> There is a large and constantly growing list of alloy compositions which meet these requirements, and it has not been deemed advisable to infer any limitation

by including some few of them in Table 40. Depending on composition and heat treatment, some "carbon steels" meet the requirements for a considerable range of temperature. In general, the 3-4 per cent nickel steels are acceptable for quite low temperatures, and the 18 per cent chrome —8 per cent nickel stainless steels meet requirements to very low temperatures. Various other classes of alloys also are acceptable. Apparently minor variations in composition and heat treatment make large differences in toughness at low temperatures.

A rule is given for figuring *S* values for unlisted materials, a table of joint efficiencies is given, and it is required that weld metal meet the above impact requirements.

These requirements are not as complete as might be desired as regards number of specimens per test, number of tests required or method of interpreting the results. However, the present state of the art of impact testing recognizes the value of the test for indicating materials which are, through their inherently brittle nature at low temperatures, unsafe for low temperature service, without attempting to specify too definitely items which are still the subject of much dispute. This section of the code, therefore, recognizes the value of the impact test without being unduly restrictive as to details which are not yet sufficiently crystallized to permit inclusion in a basic safety code.

Design Pressures

Design pressure for high and low sides for 16 different refrigerants are specified, being two-thirds of the test pressure given in a table taken from the B9 code. Rules are given for determining design pressures for refrigerants not listed. The rule for figuring pipe thickness is the same as in other sections of the pressure piping code, but this is supplemented by various special minimum requirements carried over from the B9 code and applicable to refrigerants only.

This section has included, as in the preceding code sections, the requirement that in computing the wall thickness of pipe, the service pressure assumed in the formula shall not be taken as less than 100 psi. This is to preclude having pipe wall thicknesses so thin that, although capable of withstanding the internal pressure, the pipe would be unable to stand rough handling, or mechanical strains encountered in service such as would result from stepping on the pipe, or from having to bear the weight of a ladder and workman. This does not conflict with the statement that the design pressure for the piping systems shall be two-thirds of the test pressure, even though this results, in many cases, in a design pressure considerably below 100 psi. The 100 psi restriction applies only to the pipe or tubing used in the system, and if lower pressure fittings, for instance, are suitable for the design pressure, they would be permitted, if satisfactory for the service.

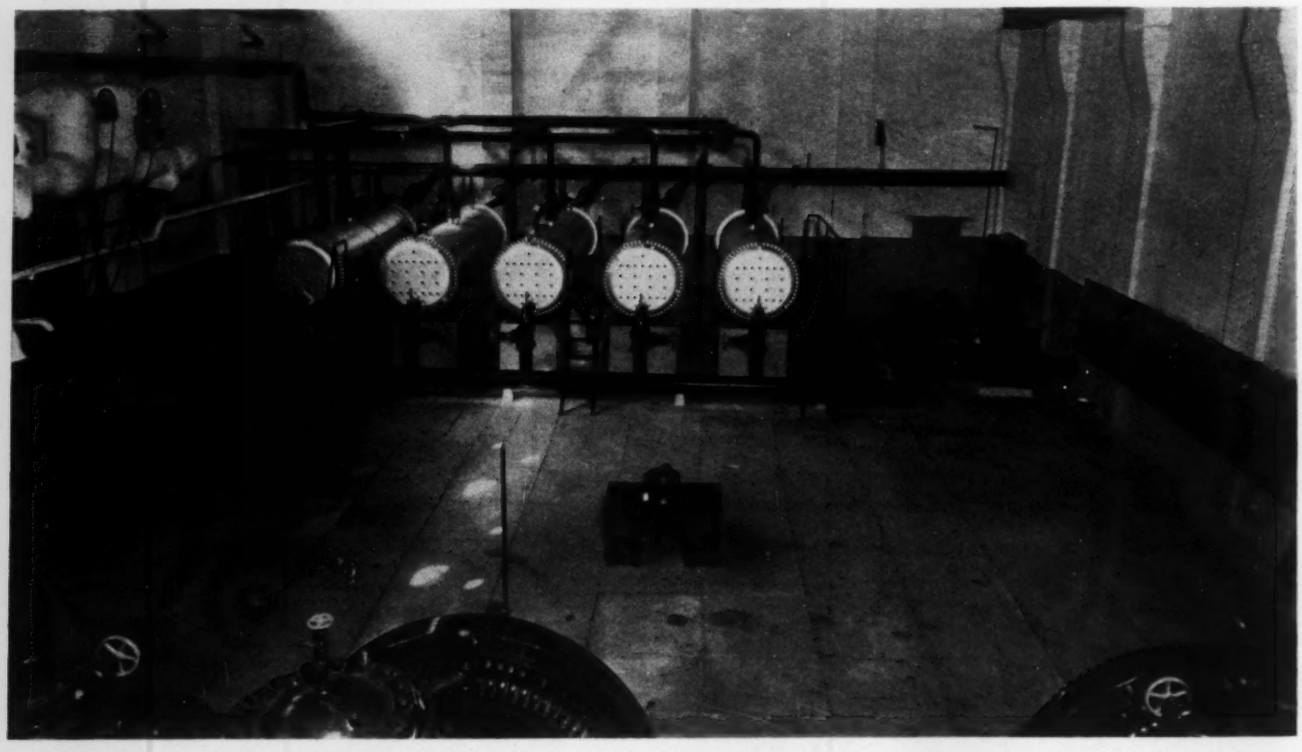
Pipe

Since the corrosion factor for plain end non-ferrous pipe or tubing is designated as zero, it is quite possible to secure extremely thin walls, particularly in the small sizes. To provide for adequate mechanical strength in this case, certain minimum wall thicknesses have been specified as follows:

For plain end non-ferrous pipe or tubing, minimum wall thicknesses shall be as follows: for nominal sizes up to 1 in., the nominal thickness for field erection shall not be less than specified for type L, and for factory assembly not less than specified for type M in ASTM specification B88; for nominal sizes 1 in. and larger, the nominal wall thickness shall in no case be less than 0.049 in. Additional wall thickness shall be provided as required to care for corrosion, erosion or mechanical strength.

Various special requirements as to location of refrigerant pipe, its

A large ammonia condenser installation, showing ammonia piping above and water line below. The compressors are in the foreground. The insulated suction piping may be seen at the left



protection from mechanical damage, the locations where hard or soft solder can be used and other items, carried over from the B9 code, are incorporated.

Pipe Joints

Joints for non-ferrous piping may be screwed, flanged, soldered (sweated capillary) or of the flared compression type, provided restrictions on sizes for threading and for dimensions and materials are followed. For refrigerant pipe, soldered joints in pipe or tubing erected on the premises must be capable of withstanding a pull apart test equivalent to an internal pressure of not less than 300 psi with a temperature of 800 F. In systems containing 20 lb of refrigerant or less, however, this requirement does not apply for pipe and tubing ½ in. nominal size or smaller. Flared compression fittings of an approved type such as specified in the American standard for brass fittings for flared copper tubes, ASA A40.2, are permitted for joints on annealed copper tubing not exceeding ¾ in. outside diameter, *provided that all such fittings shall be exposed for visual inspection.*

Pipe joints for steel and wrought iron pipe may be screwed, flanged or welded. Threads for screwed joints must conform to the American standard for tapered threads, ASA B2, and welded joints must conform to the detail requirements contained in section 6, on fabrication details, of the code for pressure piping.

With further reference to pipe joints, another requirement peculiar to this section which has occasioned much discussion is that bushings shall not be used in fittings unless the reduction is two or more pipe sizes. For single pipe size reduction, reducing fittings shall be used. While bushings for single pipe size reductions are made and are in current use for other services it was the consensus of the committee that this practice was not suited to refrigeration piping and should not be permitted in a safety code.

Stop Valves

For refrigerants, stop valves are required to be of the globe, angle or needle type. Gate and other non-globe type valves and cocks shall not be used with refrigerants except in such industrial plants as are completely separated from residential, business and other built-up sections and from public buildings and playgrounds, and which have competent operating men constantly in charge. In such locations, gate and other non-globe types of valves and also constantly lubricated cocks are permitted for vapor and gas pipes of systems using the hydrocarbons or ammonia as the refrigerant.

All refrigerant valves having stuffing boxes or valve stem packing must be of the backseating type to permit repacking under pressure. On systems containing more than 20 lb of refrigerant, stop valves are required at each inlet and each outlet pipe of each compressor and at each outlet of each liquid receiver; in addition, on systems containing 100 lb or more of refrigerant, stop valves are required at each inlet of each liquid receiver and at each liquid and each suction branch header. Stop valves used with soft annealed copper tubing or hard drawn copper tubing ¾ in. nominal size or smaller must be securely mounted independent of tubing fastenings or supports. Stop valves placed where it is not obvious what they control must be suitably labelled.

Tests

Adequate test requirements are included to demonstrate that a tight system has been secured. Every refrigerant containing part of every system shall be tested and proved tight at the designated test pressure after complete installation. Equipment such as compressors, control mechanisms, etc., which is factory tested is not embraced by this requirement since it is outside the scope of this code.

Specific Requirements

Specific requirements cover primarily the prohibition of certain materials with certain refrigerants which react chemically with them—for example, copper and its alloys with ammonia, aluminum with certain of the "Freon" group, natural rubber gaskets with "Freon" refrigerants, etc. In addition, a general word of caution is added as follows:

It must be understood that many of the refrigerants are corrosive to the usual materials, when moisture and/or air is present, and it is assumed in approving these materials that the systems will be charged and operated in accordance with accepted practice to prevent or minimize this corrosion. Similarly, many brines are unduly corrosive to the usual materials unless the pH value is properly controlled and/or the proper corrosion inhibitors are used, and it is assumed in approving these materials that the systems will be controlled to prevent or minimize corrosion in accordance with accepted practice.

Although the refrigeration section is new to this piping code, the incorporation of the B9 provisions gives this section the same time-tested quality as the other sections, B9 having been issued as an American standard first in 1930, and having been revised in 1939 after an extended and painstaking discussion. The respective jurisdiction of the two codes remains to be worked out, but since there are no conflicts between the two, and there is a considerable overlapping of personnel, this should not prove difficult. The piping code section is the more comprehensive of the two in its coverage of piping.

FLOW METERS REVEAL WASTE OF STEAM

The flow meter chart showing the 24 hr record of steam consumption in the hog killing department of a Chicago plant indicated consumption of steam from 6 p. m. to 6 a. m., when the department should not have used steam.

Investigation disclosed that steam had been permitted to leak into the hot water heater all night, some one having neglected to close the steam valve tightly. The loss was 111,600 lb of steam.

This is but one illustration of the value of departmental steam flow meters.—S. D. Distelhorst, Cochrane Corp.

For news about new and improved equipment, see Equipment Developments in HPAC every month

The Man Behind the Man Behind the Man Behind the Gun—the Air Conditioning Engineer

PRODUCTION workers busy 24 hours a day manufacturing war material for our fighting forces need the services of the air conditioning engineer in providing suitable atmospheric conditions for efficiency. In machine shops and other factory areas where heat gain from motors and other processes is heavy, where sun heat pours in through immense roof areas, where tolerance requirements are exacting, air conditioning in some one or more of its forms is essential. In plant office buildings and engineering departments, too, perspiration and discomfort menace efficiency. Therefore, the air conditioning engineer may often be called "the man behind the man behind the man behind the gun" . . . F. O. Jordan, of Albert Kahn Associated Architects and Engineers, tells of some of the air conditioning problems that arise in connection with the huge new war plants, explains their solution in two recent cases

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WHEN THE aviation industry of America was forced to expand its manufacturing quarters in order to keep pace with the demand for military airplanes, it quite naturally looked about for the latest developments in construction and equipment of industrial buildings. Regardless of scientific production equipment and high pressure production schedules, production cannot be maintained at its peak—nor products at their highest quality—unless the comfort and health of the workers are considered. Since this is a function of air conditioning—and since air conditioning is vital to many of the processing operations—air conditioning in one or more of its forms has been installed in practically all of the large new plants.

In machine shops and similar areas where the heat gain from motors and processes is very heavy and where tolerance requirements are exacting, complete all year control of temperature and humidity, usually involving mechanical refrigeration in the summer, has often been installed. Similar comfort control is installed in plant office buildings and engineering departments where perspiration and discomfort menace efficiency. In buildings where subassembly and assembly of airplane structures is carried on, occupancy and motor loads are much lighter and tolerance requirements less exacting. Here the practice is often to utilize outside air change or evaporative cooling for preventing atmospheric conditions from getting out of hand.

Industrial plants of large size present special problems when they are air conditioned, both in their design and installation of the systems. The principal load factors are sun effect, motor loads, ventilation loads, and in some cases, illumination and occupancy. The enormous roof areas and unusual ceiling heights also present problems all their own. A description of two installations—and some of the problems involved—will serve to illustrate methods of cooling large industrial plants not only for the aviation industry but for others engaged in war work.

The main factory building at one aircraft engine plant is in reality a huge machine shop. The building is divided by means of a masonry partition into an assembly portion and the machine shop proper.

The following data were among those used in calculating the cooling load:

Outside conditions: 98 F dry bulb, 78 F wet bulb, 85 F effective temperature, 70 F dew point.

Average inside conditions: 82 F dry bulb, 68 F wet bulb, 75 F effective temperature, 61 F dew point.

South monitor glass (sun resistant): 1.1 Btu per sq ft per hr per F transmission and 50 Btu per sq ft per hr sun effect.

Other monitor glass (clear): 1.1 Btu per sq ft per hr per F transmission.

South windows (sun resistant): 1.1 Btu per sq ft per hr per F transmission and 40 Btu per sq ft per hr sun effect.

Other windows (clear): 1.1 Btu per sq ft per hr per F.

Due to the high ceiling, the temperature at the breathing line will be lower than the average temperature used as the basis of calculations, while the temperature in the monitors and near the roof will be higher. The temperature in the monitors worked out to be about 90 F, so transmission loads were calculated on a 10 F differential for the roof and monitor glass and gunite. No transmission was calculated for the floor, as ground temperatures in air conditioned buildings ordinarily do not rise above the inside temperature. Transmission for walls

and windows was calculated on a basis of a 16 F differential.

Sun effect was calculated for the south monitor glass, south windows and the roof surfaces only, as the maximum simultaneous or peak sun load occurs on these surfaces. For other exposures, transmission only was calculated.

6000 Tons

The total maximum simultaneous or peak refrigeration load for which the equipment was selected and the distribution system designed worked out at almost exactly 5000 tons of refrigerating effect for the assembly and shop. Another 1000 tons is carried by this system for air conditioning the office building and the factory offices and cafeterias, bringing the grand total for the installation to 6000 tons.

As a factor in reducing the operating cost of this huge air conditioning plant, several thousand square feet of sun screen was installed on the office windows. After the installation, it was found that the glare from the direct sunlight was materially reduced, so this added benefit may be combined with the estimated load reduction of around 200 tons for the offices alone.

Several Schemes

In addition to making the load analysis, a complete study was made for the purpose of selecting the type of air conditioning plant best suited to this particular installation. Tentative designs were made and cost estimates prepared for a system for cooling by pumping outside air through the building, a system using evaporative cooling units taking in all outside air, and air conditioning systems using refrigeration. Considering the heavy loads on the plant and the necessity of maintaining production volume and product quality throughout the year regardless of the weather, the management decided in favor of control of temperature and humidity by means of a refrigeration plant.

Actual bids were received covering direct expansion and indirect systems. Under indirect systems, bids on both centrifugal machines and steam jet equipment were submitted. These bids showed the indirect system with steam jet equipment to have the lowest first cost, with the indirect system using centrifugal compressors second and the direct expansion system highest. When total annual costs, including fixed charges and operating costs were estimated, the centrifugal machines were estimated to be the most economical.

Of course this would not always be the case. In locations where an ample supply of satisfactory condenser cooling water is available, the steam jet may show the lowest annual charge. In many cases, especially on smaller installations, the direct expansion system may show a considerably lower cost than the indirect plant. At this plant it was possible to combine the winter heating and summer cooling systems so the same water piping, heat transfer surfaces, circulating pumps, fans, air filters, and air ducts are used both for heating and cooling. Hence, all year air conditioning was available merely by adding little more than the cost of the water cooling equipment and cooling tower to the cost of the heating system.

Another factor that worked against the direct expansion system was the owner's requirement for a plant that would make use of steam from the otherwise idle steam generating plant during the summer. Thus it was necessary to include the cost of a turbogenerator with the cost of the direct expansion system to match the steam turbine driven centrifugal compressors of the indirect plant.

The factory space is air conditioned by 70 central systems situated in the overhead truss space. Each of these systems consists of a fan of approximately 36,000 cfm capacity, drawing through an automatic oil type air filter, a combination heating and cooling coil and entrained moisture eliminator plates, and discharging through about 400 ft of ductwork. Hot water is circulated through the coils in winter and chilled water in summer by means of mains situated in the truss space and steam turbine driven circulating pumps in the compressor house. The quantity of water circulated is 12,000 gpm at a head of 100 ft.

1200 Ton Units

In winter the water is heated by steam surface type instantaneous heaters, and is chilled in summer by means of five water cooling units of 1200 tons refrigerating capacity each. Each water cooling unit consists of a condensing type steam turbine with surface type condenser, driving a centrifugal compressor with surface type refrigerant condenser. Condenser cooling water is circulated through each pair of steam and refrigerant condensers in series. This water is cooled by a cooling tower with mechanical draft, the quantity of condenser cooling water being 18,000 gpm. All of this equipment, except the cooling tower, is installed in the compressor house portion of the boiler house.

Operating conditions were carefully selected to obtain the highest possible capacity and efficiency at the lowest cost. In general, higher chilled water temperatures penalize the air conditioning coils and fans because they make it necessary to install more coil surface and circulate more air to carry a given refrigeration load. However, higher chilled water temperature tends to increase the capacity and efficiency of the compressor and reduce operating costs. Likewise a greater water temperature rise through the air conditioning coil makes more coil surface and larger fans necessary, but reduces piping and pumping costs.

By delivering the chilled water from the water cooling units at 47 F and using a rather low refrigerant condenser temperature, it developed that standard production compressors with a tonnage rating lower than 1200 at 45 F water off the cooler would deliver 1200 tons. With this water entering the air conditioning coils at 48 F and leaving them at 60 F, a standard four row coil shows the low proportion of latent to sensible work required by the high sensible load on this installation.

Cooling Tower

The cooling tower was designed for an "on" temperature of 107 F and an "off" temperature of 85 F, with a wet bulb air temperature of 78 F as this resulted in an econom-

ical design. The temperature rise of 22 F through the condensers is obtained by passing the water first through the refrigerant condenser and then through the steam condenser, and by sizing the condenser tubes so that all of the water passes through them, rather than bypassing some of it around them. One of the reasons for the lower first cost of the indirect system on this particular installation is this selection of operating temperatures for maximum capacity and efficiency of *all parts* of the system.

Control Methods

The chilled water temperature in the summer time is automatically controlled at 47 F from the coolers by throttling the steam supplies to the turbines under automatic regulation of thermostats placed in the chilled water outlets from the coolers. In the winter, the hot water temperature from the heaters is automatically regulated according to the outside temperature. The compressors are provided with the conventional safety controls, such as automatic shutoffs which function in case of excessive speed, failure of oil pressure, failure of condenser cooling water supply and dangerously low chilled water temperature.

The air conditioning systems are controlled as follows:

An effective temperature thermostat is situated in the recirculated air intake for modulating the air bypass damper around the air conditioning coil toward "open" upon falling effective temperature during the cooling season and upon rising temperature in the heating season.

The air mixture temperature entering the air conditioning coils is kept above 40 F in the winter and below 85 F in the summer by modulation of the outside air and recirculating dampers under control of a dry bulb thermostat situated in the mixing chamber between the air filters and the air conditioning coil. Vent dampers are modulated automatically in conjunction with these dampers to relieve the pressure in the building due to the outside air supplied.

Automatic seasonal changeover controls are also provided. With this control system the cooling and heating capacity of each air conditioning coil is varied automatically according to the load on the particular zone served by the coil, and freezing of the coils is prevented by limiting the outside air supply—yet the outside air is not permitted to fall below the requirements for ventilation.

The proportion of outside air is increased automatically as the outside temperature rises until heat is no longer required. As the temperature continues to rise, 100 per cent outside air is introduced as long as its temperature is low enough to offset the internal heat gains in the zone. Rising inside effective temperature throws the cooling cycle into action, and a continued rise in outside temperature modulates the outside air dampers toward the minimum outside air required for ventilation in hot weather.

Thermostatically controlled unit heaters are placed at the outside walls in case any exposure requires heat during periods when cooling is demanded further inside the building by internal heat loads.

A Typical Plant

The other plant to be described here is one of several similar ones the writer assisted in engineering for another owner. In each case the office and engineering sections have complete all year air conditioning, while the factory portions are cooled by pumping outside air through them. The one described is typical of the others.

The total refrigeration load on the office and engineering sections is slightly over 850 tons. The load periods on various parts of the system vary greatly, as the dining rooms may require cooling when the rest of the building does not, or offices often will be in use after regular working hours. For these reasons the installation is divided into nine separate direct expansion systems, using reciprocating compressors and evaporative condensers. The same systems are used for heating by means of steam coils. Each system consists of air filters, direct expansion cooling coils, steam heating coils, fans, compressors and condensers situated in central fan rooms, distributing conditioned air to the conditioned spaces through concealed ducts with directional type registers.

All systems are provided with outside air and recirculating dampers modulating to maintain certain temperatures entering the coils as explained above. Refrigeration equipment as well as steam for heating are under control of thermostats in the return air ducts.

The factory building, where the roof height varies from 30 ft in the fabricating and subassembly areas to 60 ft in the assembly areas, is cooled by circulating outside air in sufficient quantity to prevent the temperature at the breathing line from rising more than 5 or 10 F above the existing outside temperature.

The outside air is introduced by mechanical type ventilators distributed over the entire roof area, taking the air in from above the 6 ft level. Tests have shown that air at this height above the roof is not greatly affected by the radiant heat from the roof surface. Discharge ducts are extended downward from these ventilators to the bottom chords of the roof trusses to avoid interference with the natural tendency of the hot air to stratify near the roof where it does not affect the temperature of the breathing line. Nozzles are provided at the bottoms of these ducts for directing the air at an angle toward the floor at about 2000 fpm velocity so that convection cooling is obtained.

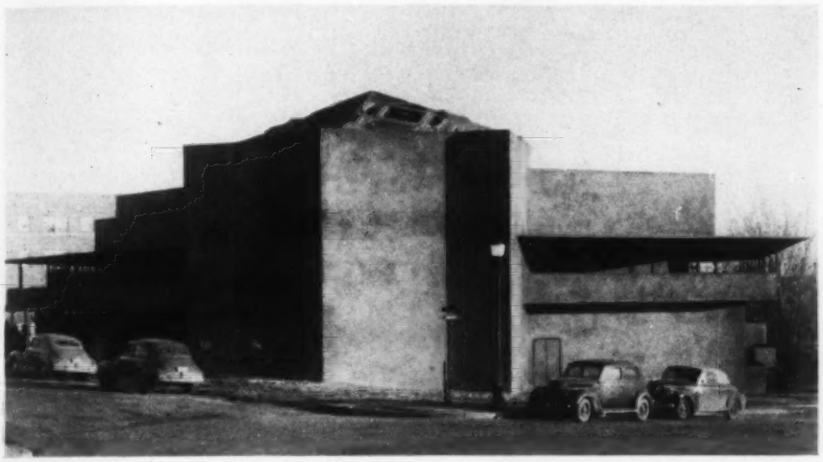
The hot stratum of air is pulled off from the under side of the roof by mechanical exhaust ventilators placed at the high points of the monitors.

About 2 cfm per sq ft of floor area is required for this plant. All ventilating units are provided with 36 in. propeller fans with a capacity of 10,000 cfm each, so that one supply and one exhaust unit are provided for each 5000 sq ft of floor. All of the ventilators are weatherproof and lightproof.

In a building of this type, the temperature close under the roof at which the air is exhausted is about 20 F above the outside temperature at which it is introduced. The air quantity may readily be calculated by dividing the heat absorbing capacity of air at a 20 F temperature rise into the heat gain due to sun effect and internal heat producing equipment.

Floor Coils Heat Auditorium of New Community Church

Wrought Iron Piping Embedded in Floor Construction in Unusual Structure



Designed by Frank Lloyd Wright, the new Community church in Kansas City has many unusual features. Panel heating is used in the auditorium, and it is planned to circulate chilled water through the floor coils for precooling during the summertime

FRANK LLOYD WRIGHT prepared the plans for the new thin-walled steel and gunite building of the Community church in Kansas City, which is a structure apart from its fellows. From its heating and air conditioning system (designed and installed by Westerlin and Campbell) to its flat roof and shallow crown from which searchlight beams eventually will slice the heavens with a spire of light, it is different. Seating arrangements are similar to those of an amphitheater. Rows of theater seats are on steps that rise from the pulpit platform to the rear of the auditorium. The aisles are gently sloping ramps.

In this church, gunite comes into its own as a primary building material. Shot under pressure, it has formed the walls, ceilings and floors. The walls are about 3 in. thick. They were built about a "bird cage" of 2 in. square steel tubing. Onto that went paperbacked mesh and then the layers of gunite, leaving a 2 in. insulating air space between the inner and outer walls. B. E. Wiltscheck was the general contractor.

This church has no windows, except those in the offices. At the present time there still are two units to be built, a chapel and auditorium to seat about 250 persons, a kitchen, dining rooms and additional classrooms; and extensive parking levels and ramps. Because of the war, those units will remain in the plan stage for the present.

Floor Heating

Beneath the floor of the 1200 seat auditorium, with its upholstered individual theater chairs, run coils of welded wrought iron pipe. The piping is spaced on the unit lines of the building. The supply pipes pass beneath the center of the seats, the cooler return pipes beneath the feet of the worshippers.

The auditorium of the church is heated by this floor panel, which consists of 3600 ft of 2 in. welded wrought iron pipe coils on 16½ in. centers connected to 2½ in. supply and return headers in the aisles. As the 2 in. coils are slightly pitched on each step, the pipe is at varying depths in a 5 in. layer of crushed rock that is topped with less than an inch of sand, reinforcing and the 1 in. gunite flooring.

The system used is ordinarily referred to as "radiant heating," but the term is a misnomer, Mr. Wright maintains. "Gravity heat" is precise and correct, he says, and points out that heat rises as naturally as water falls. Any heat generated beneath seeks to rise above. This heating system is therefore natural and economical, he feels.

The balcony, which forms a part of the main auditorium, has no individual heating element, but is heated mainly by the rising warm air from the floor area of the auditorium. Since the temperature variation from floor to ceiling with a system of this type is low, the temperature of the air in the balcony does not vary much from that at head level on the main floor of the auditorium. Since there are no outside walls to the balcony, there is less need for separate heating elements.

Ventilation of the auditorium is accomplished by exhausting stale air through two sets of exhaust grilles in the ceiling at both ends and above the balcony. New air is introduced through infiltration and through the air ducts on each of the forced air units.

Floor Cooling

The floor heating system for the auditorium was designed to permit circulation of chilled water through the coils during the summer. As the floor piping is designed primarily as a heating medium and works through a range of about 20 F (180 supply, 160 F return), it is obvious that the quantity of pipe coil from a cooling standpoint would be entirely inadequate to take care

SUMMARY—The auditorium of the new Community church in Kansas City is heated by means of welded wrought iron pipe coils embedded in the floor, through which hot water supplied by an oil burning boiler is circulated. In summer, chilled water is to be circulated through these floor coils for precooling. . . . For other rooms in the structure, there are factory built air conditioning units, served with hot water in winter, and with chilled water in summer from a 30 hp condensing unit. . . . The editors acknowledge with thanks the cooperation of V. Walters, Westerlin and Campbell Co., S. A. Musser, A. M. Byers Co., and Sam Smith, Kansas City newspaperman, in furnishing information for this description

of the entire load in the auditorium. Therefore, it is planned to circulate 85 gpm of 53 F water through the auditorium floor coils only prior to occupancy, during which time these coils will have sufficient capacity to take care of the sensible heat load of the auditorium. It is anticipated that chilled water will be circulated through the coils for several hours prior to occupancy to do a partial precooling job, and the floor piping system is provided with a valve that immediately cuts out when any of the zone air conditioning units are started. At that time, the full refrigeration water cooling equipment is available for the three zone air conditioning units and chilled water cannot be circulated through the floor.

There are a number of questions in regard to the cooling system that only the experiences of this coming summer's operation can answer. Will there be enough cooling in the auditorium by precooling to maintain relative comfort for the period in which the congregation is in the church? Will there be condensation on the floors? How many hours before the church sessions must the refrigeration unit be started?

If the experiences on these points are satisfactory, a new field for the dual use of embedded coils will be opened—radiant heating and "radiant" cooling.

Air Conditioning

There are three zone air conditioning units; the operating schedule of the building does not necessitate the operation of all three at one time. The system is therefore designed in such a manner that when zones 1 and 3 are operated, zone 2 is inoperative. However, during the winter heating cycle all of the air conditioning units, together with the floor piping system, are in operation. Zones 1 and 3 comprise rooms in the basement and first floor and some second floor classrooms, and zone 2 consists of other classrooms and the foyer on the second floor.

Virtually all of the air conditioned spaces served by the units open into the auditorium itself and consequently it is anticipated that with these units in operation, they will have some cooling effect on the auditorium.

Due to this zoning arrangement, the 30 hp "Freon" condensing unit is actually serving areas that have a much greater load. However, the design of the control system is such that at no time will the capacity of this unit be exceeded during its operating period. It is planned to maintain 85 F dry bulb with not to exceed 50 per cent relative humidity in the conditioned spaces when outside conditions are 100 F dry bulb and 78 F wet bulb.

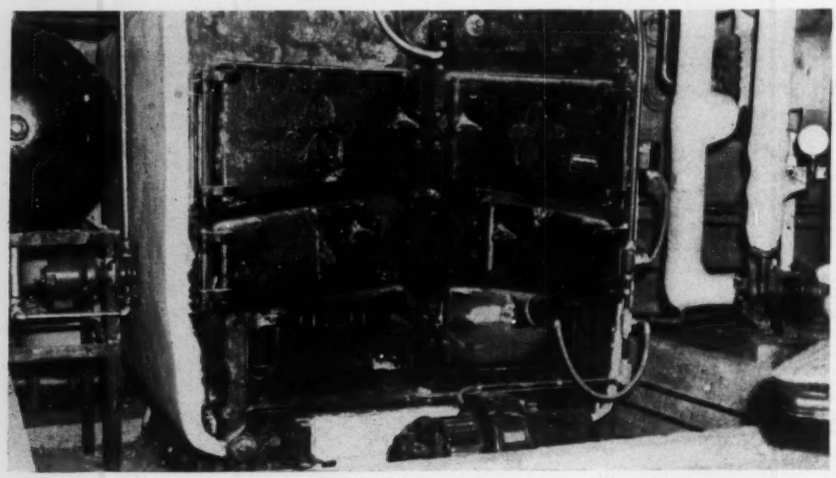
The auditorium is heated by a floor panel made up of 3600 ft of 2 in. pipe coils

The heating plant consists of an oil fired water tube boiler having 7800 sq ft of radiation capacity. A centrifugal hot water pump driven by a 3 hp motor is used to supply hot water to the floor panel in the auditorium, the three forced air units and the radiators in the lavatories.

The basement air conditioning units serving zones 1 and 3 discharge directly into furred vertical shafts which open into inter-beam plenum spaces at each floor level. The floor construction consists of 12 in. I beams on 5 ft centers with a 1½ in. gunite floor slab and a ½ in. gunite ceiling.

The original design provided for the use of open web steel joists, which would have provided for continuous plenum areas in each ceiling throughout the building. However, the building department insisted that solid web steel joists be used, which necessitated using certain spaces between steel beams as air distributing plenums; occasionally, in order to reach some remote location, vents were cut into the webs of the beams to supply air from one inter-beam space to another; baffles are provided to limit the flow of air to a predetermined direction. In other words, the actual steel construction itself becomes the duct distributing system, and the air flow into each conditioned area is handled through rectangular ceiling air diffusers.

The floor heating coils in the auditorium and the heating coils in the air conditioning units are supplied with hot water from this oil burning boiler



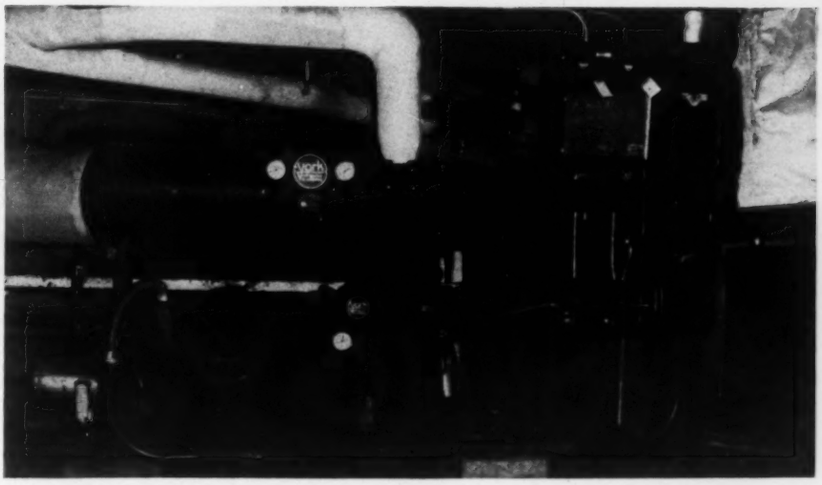
Venting the rooms was accomplished by undercutting the doors 1½ in.

The air conditioning unit for supplying the circulating air for the second floor foyer and other classrooms is in a penthouse.

Control Methods

The primary control for the heating system is the water temperature control on the boiler. By this control high and low limits on water temperature can be set. While there is no manner by which water of different temperatures can be supplied to the floor coils and each of the forced air units, there is a compressed air operated valve for each circuit. These valves are actuated by the thermostats in the respective zones so that, as the thermostats of the respective zones become satisfied, the valves are closed. The heating systems were designed for a 60 F air temperature in the auditorium where the panel or radiant heating method is used, and 70 F where the forced air units are used.

Each air conditioning unit is furnished with an automatic outside air damper and an automatic return air damper, controlled by a dual type thermostat placed in the mixing plenum ahead of the heating coils. Each outside air damper is provided with a gradual switch for positioning the opening of the outside air damper during both the heating and cooling cycles. The outside air dampers also are under the influence of an immersion type thermostat situated in the water leaving the heating coils and arranged to close the outside damper in the event the water temperature becomes dangerously low.

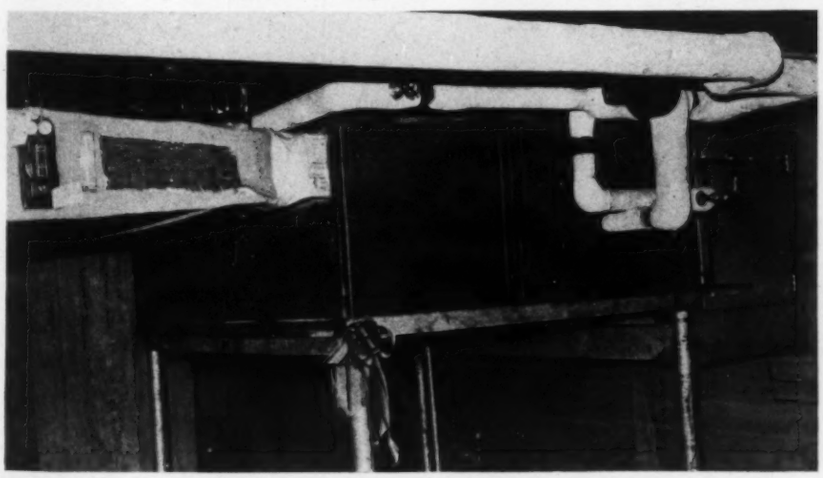


A 30 hp condensing unit supplies chilled water for summer air conditioning

In normal winter weather in Kansas City the system heats the building satisfactorily in about two hours. On the dedication Sunday, with the mercury hovering near zero, however, a full house shivered in overcoats.

It is unfortunate for a new church and a new heating system that there was not sufficient heat on the dedication Sunday. The engineers and contractors for the heating and cooling systems had not had an opportunity for checking and balancing them. It was found that the hot water pump had been running backward, due to switched electrical phases. What was more important, the valve for the floor coils was closed inasmuch as the thermostat controlling the valve was satisfied. There were two reasons for the panel system thermostat being satisfied: (1) it was in the rear of the auditorium where it was affected more by one of the forced air units than by the floor system; (2) the wall to which the thermostat was attached was unusually warm on account of a duct passing through it. With some alterations, this difficulty has been partly overcome.

One of the factory built air conditioning units which serve various rooms in the church



EXTENSION OF RATINGS BY CONTRACTORS FACILITATED

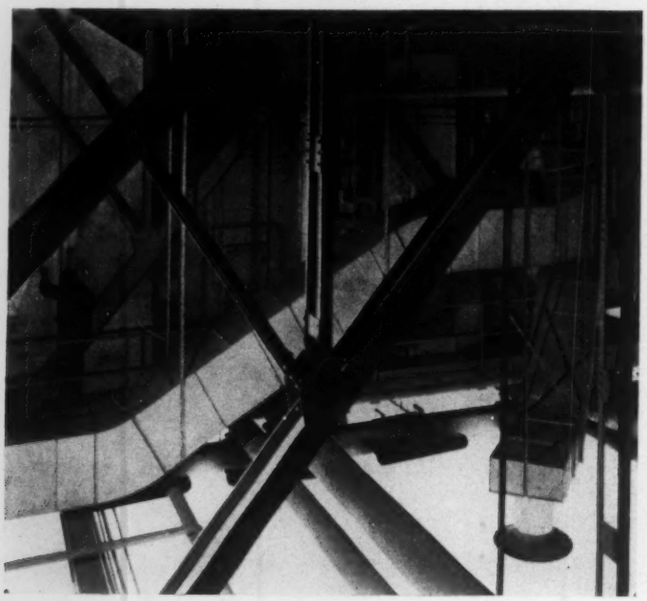
Extension of preference ratings by building contractors is facilitated by a new form of order, P-19-h, used to assign ratings to construction projects in response to applications on forms PD-200 and PD-200A, the WPB announced last month.

The new form of order will permit application of ratings by builders by a simple endorsement on purchase orders, containing the serial number issued for the project. A similar form of endorsement may be used for extension of ratings on orders served on the builders' suppliers, except that before extending a rating a supplier must execute and forward to the War Production Board a special form of acceptance. Only one filing of the acceptance is necessary, after which the supplier may extend any rating assigned by order P-19-h, regardless of serial number.

Suppliers to whom ratings are extended under the new order may accumulate the ratings up to a period of three months until they can place an order for a minimum commercial quantity of the rated material if they do not process it in any way, and, subject to the same restrictions, suppliers may "basket" or accumulate ratings on orders from two or more contractors.

All photos courtesy the Austin Co.

AIR CONDITIONING — FOR OFFENSE — ASSURES UNINTERRUPTED PRODUCTION OF WAR MATERIAL



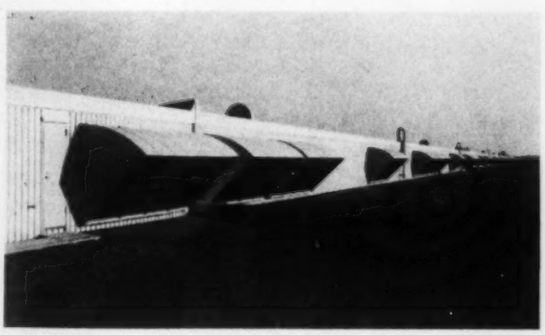
ABOVE

Raising two large fans into position on one of the air conditioning platforms on trusses above the main assembly aisle in a windowless, controlled conditions manufacturing plant recently completed by the Austin Co., engineers and builders, for the production of war equipment. The plant is air conditioned because of the need for uninterrupted production at the fastest possible rate through the sultriest summer heat

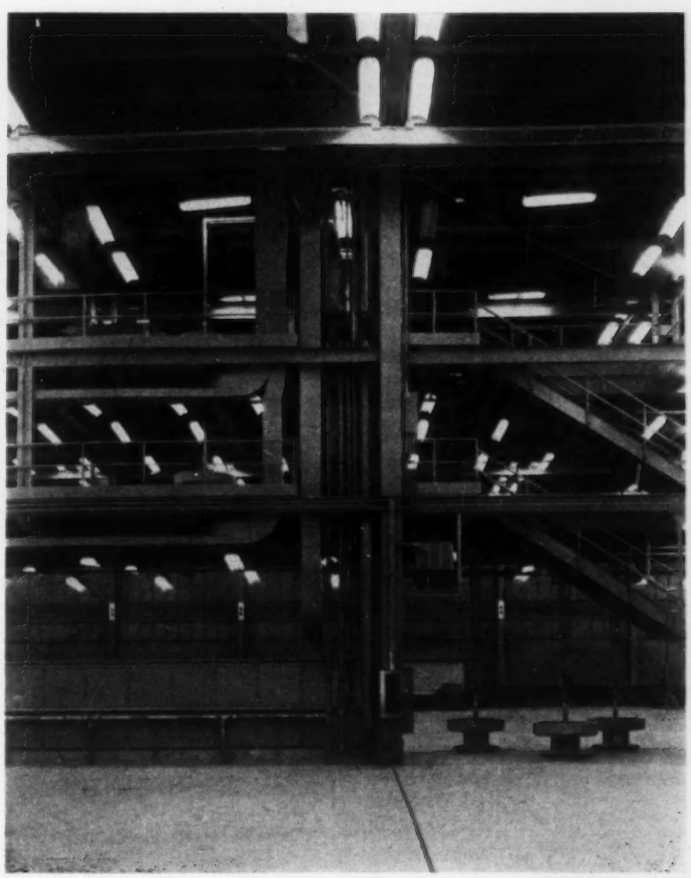
AT LEFT

A typical air conditioning platform high above the working area, supported by trusses that span the big plant's assembly bay. Ducts reaching out in four directions from each unit serve an area of more than 75,000 sq ft. Chilled water is circulated to the air conditioning units from steam turbine driven refrigerating compressors. For the heating cycle, steam from the gas-fired boilers will be used in interchangers to heat water which will be circulated through the same piping which conveys the cooled water during the summer. The boilers have an emergency fuel oil system

Special booster coils installed in air conditioning ducts above door openings and sections reaching the building's outer extremities will expedite temperature control throughout the windowless manufacturing plant, where fiber glass and steel have been used in shatterproof side walls, heavily insulated for air conditioning economy.



Outside air is taken in and used air is exhausted through these hoods, which are equipped with special light traps to complete the blackout protection of the plant. The difference in roof level between the section from which this view was taken and the higher section from which the hoods extend is accounted for in the greater depth of trusses in the plant's assembly aisle, above which all air conditioning equipment is situated



AT RIGHT

Toilets, wash fountains and all other convenience facilities in the plant are placed at frequent intervals under or on the mezzanines, directly adjacent to the assembly aisle. Special ventilation is provided by means of fans which serve ducts leading from these areas direct to the roof. Note the general absence of columns in this area, which has been kept free by suspending mezzanines and stairs from above

"OPEN FOR DISCUSSION"
Communications to the Editor

PROTECTING STEEL HEATING AND COOLING COILS

THE EDITOR—

I have read the article on How Does the Switch to Steel Affect Heating Surface Performance? published in the War Problems Forum of the March HPAC.

During the heating season, there should be no trouble from corrosion of steel surfaces of heating coils after air washers, in my opinion. But in the summer season, if steel coils are used, there will be a rapid corrosion attack, as shown by the paper by W. Z. Friend in the ASHVE Journal Section of the March HPAC.

We would advise that heating coils have heavy zinc coatings, either plated or sprayed on by means of the Schoop process. Another means of protecting these pipes is by a zinc chromate primer, in a 100 per cent phenolic resin vehicle, followed by two finish coats of zinc dust, zinc oxide pigment, in a 100 per cent phenolic resin vehicle. The diminution of heat transfer rate will be very slight.

The same procedure should be used for cooling coils. Here, however, the problem is one of condensate picked out of the air in the cooling season. This condensate is, in effect, distilled water which picks up the soluble acid gases in the air, thereby making the condensate very corrosive.

It is doubtful if galvanizing would stand up for any length of time under the acid distilled water attack just mentioned. It is quite probable that the painting with the phenolic resins would prove more effective. It is acknowledged that in this case, the diminution of heat transfer would probably be greater than in the case of the heating coils, due to the fact that there is a much larger temperature differential between the steam in the coils and the air, than against chilled water (or refrigerant) and the air.

In connection with water treatment, as mentioned in this article, I think it very important that steam containing as little CO_2 as possible be supplied to the coils. Heating the feedwater to above the boiling point in a venting heater should remove practically all the CO_2. If this cannot be done, then treatment of the boiler water with some of the newly developed amines should change the CO_2 to a complex ammonium carbonate and at the same time raise the pH value of the condensate.

It is important that pH studies of the condensate be made with glass electrode potentiometers and that the condensate be taken through a bottle chain to prevent contamination by air. Determining the pH value of a condensate or any distilled water by colorimetric methods is not accurate as the condensate tends to take the pH value of the particular indicator used. I have seen nine different pH values obtained from the same condensate, using different indicators, in acid and alkaline ranges—C. M. STERNE, chief engineer, Metropolitan Refining Co., Inc.

HIGH VELOCITY AIR CIRCULATION IN ARMORIES

THE EDITOR—

My attention has been directed toward a statement appearing in the January, 1942, issue concerning the first non-industrial application of the circulation of air at high velocities for heating and ventilating a large gymnasium and armory floor as applied to the new Patten Memorial gymnasium of Northwestern university, Evanston, Ill.

I take exception to this statement because in 1937 when I was mechanical engineer for the board of public service of the city of St. Louis, under Wm. C. E. Becker, division engineer, I designed and supervised the installation of a system of this kind for the new armory in St. Louis. To the best of my knowledge and because of remarks made by contractors bidding on the work, this was the first system utilizing the above method that was planned for the St. Louis area.

The origin of this method of heating and ventilating is not mine, however; I heard of it through the St. Louis representative of the B. F. Sturtevant Co., who introduced me to his factory manager who was in St. Louis at the time and who was familiar with this method, which was then used with excellent results in what were then new armories located in New York and New Jersey.—JOHN C. ROSS, mechanical engineer, St. Louis board of education.

Providing heat for a building under the following weather conditions caused heating engineers to wrinkle their brows recently, says R. C. Lindblom, of the General Electric Co. At the site of the building the average wind velocity from December to March is about 55 mph, with a recorded high of 231 mph. The mean temperature runs from about 5 F in January and February to 45 F in the summer months. The lowest temperature in the last eight years is 46 F below zero and the highest temperature reached in the summer is 71 F. Snowfall averages more than 200 in. a year. . . . These data are precise because the new building, housing station W39B, new FM transmitter of the Yankee network, is on the top of Mt. Washington, in New Hampshire, right next to the U. S. weather bureau station. . . . Two oil fired winter air conditioners with a total rated capacity of 400,000 Btu per hr handle the load. With the two units, there will always be enough heat available to keep all parts of the building comfortable, even if one unit should shut down for a short time. At first, use of a single 250,000 Btu unit was considered.

WAR PROBLEMS FORUM

HOW TO SAVE CRITICAL METALS IN SPECIFYING NONSPARK FANS

The problem of collecting, conveying and exhausting air containing a mixture of alcohol or other high volatile chemicals used in processing in industry has long been recognized as one requiring equipment incorporating special provisions for the avoidance of a frictional spark, due to the hazard this would be.

In the majority of cases, commercially made fans are used for such duty, equipped with a nonferrous wheel. Ordinarily, such wheels are fabricated of copper, aluminum or other material having a nonsparking characteristic that will prevent the generation of frictional sparks should the wheel, while revolving, come in contact with the sheet steel fan housing. For some special requirements the wheels are sometimes given a special rubber coating to provide a further safeguard against the spark hazard.

It should be stated, however, in fairness to the recognized skill of fan manufacturers, that shifting of the fan wheel toward the inlet side of the fan housing, sufficient to produce an emery wheel effect contact, very seldom occurs. Nevertheless, good judgment dictates the use of the nonspark type fan, which has become a generally accepted prerequisite along with explosionproof motors whenever dealing with a volatile contained air having or even approaching an explosive mixture.

Such fans were readily obtainable until quite recently. Because of the scarcity of metals—particularly copper and aluminum, which must be conserved for use by our armed forces and our war offense program—it seems reasonable that we should explore every possible means toward reducing the need for such critical metals.

In place of the generally accepted, standard, nonspark complete fan wheel assembly, which is not only difficult to procure but is also higher in cost, we have recently specified the use of a standard sheet steel fan and housing for this duty. A copper or aluminum ring, ⅛ in. thick, is secured to the inside surface of the fan inlet sheet. Should the wheel ever shift sufficiently to make a contact it would strike this ring rather than the steel sheet. The weight of nonspark material thus used is but a small percentage of that required when fabricating the entire wheel of copper or aluminum.—E. P. HECKEL, E. P. Heckel & Associates, consulting engineers; member of HPAC's board of consulting and contributing editors.

WPB ISSUES ORDER ASSIGNING A-10 RATING FOR HEATING REPAIRS

Issuance of an order designed to facilitate the maintenance and repair of existing heating and plumbing installations in office, apartment, farm and residence buildings, was announced March 14 by J. S. Knowlson, director of industry operations of WPB.

The order (P-84) assigns an A-10 preference rating, which may be applied by an installer or supplier, to materials needed for emergency heating and plumbing repairs. The rating may not be applied, however to obtain copper already fabricated in sheets, wires, rods or tubes or to any scarce materials which can be eliminated by change of design or by substitution.

W. Walter Timmis, chief of the plumbing and heating branch of WPB, said:

The importance of maintaining existing plumbing and heating facilities is recognized in this order, but because the plumbing and heating industry is chiefly a metal consuming industry, every effort must be put forth to keep consumption of critical materials at the irreducible minimum.

In other words, this order is not a guarantee that existing plumbing and heating facilities will be maintained in exactly their present state.

In normal times it has often been found that it is cheaper to replace than to repair. Replacement now can be made only if there is no possibility of repair. In cases

The refrigeration and air conditioning branch of the War Production Board last month asked all stores, restaurants, factories, theaters, studios, laboratories and film exchanges to ship back all empty "Freon" cylinders to their manufacturers so that they may be refilled and used again for air conditioning.

The cylinders are made out of pressed steel, a critical material. No new cylinders are obtainable and the manufacturers of "Freon" must depend on cylinders already made to pack their entire supply. The WPB believes that almost every air conditioned establishment in the country has some empty cylinders on its premises.

The importance of returning empty refrigerant cylinders promptly was stressed in the January HPAC, and suggestions for conservation of refrigerants were presented.

where replacement is essential it will be found that the items now being produced are severely limited in number of types and styles, that the weight has been reduced, and that in many cases, less critical materials are being used in place of the customary materials.

The intent of this order is not only to insure maintenance and repair, but also to insure that the products made available for that purpose are used only for *essential* maintenance and repair.

Heating equipment is defined as any primary heating unit used to provide building warmth or any accessories of such unit.

Emergency repairs are defined as those required by actual or imminent breakdown of heating or plumbing equipment. They include the emergency replacement of equipment which is worn out or damaged beyond repair. They do not include installation of superior type equipment or substitutions more extensive than required to replace worn out parts.

Other provisions of the order are as follows:

1) The A-10 rating may be applied by a simple indorsement on purchase orders or contracts.

2) Installers may not apply the rating to obtain delivery of materials on earlier dates than required to enable them to make emergency repairs.

3) A supplier may not apply the rating if he can make his delivery and still maintain a minimum working inventory.

4) The A-10 rating is applied automatically in the case of delivery to an installer of any heating and plumbing item, the cost of which is less than $5, provided that the total order placed by the installer is less than $10.

["Such deliveries," Mr. Timmis stated, "to which the rating is automatically supplied must be actual and *bona fide* purchases for less than $5. Similarly the overall order must be for less than $10. Larger orders may not be broken down so as to bring them, by subterfuge, within the terms of this provision."]

5) In the case of units costing more than $50, and worn out or damaged beyond repair, the installer must attach to the purchase order a statement signed by him and by the ultimate user, certifying that repair is impracticable, and specifying the points at which the unit failed.

6) Provisions of priorities regulation No. 1 are made applicable.

Order P-84 is designed to fill a gap in the application of orders currently existing for repair and maintenance needs. Preference rating order P-100 for instance, assigns an A-10 rating for repairs and maintenance items but may not be used to secure repairs for retail or residential properties.

The new order, although making the A-10 rating extendable to all suppliers, including the manufacturer, does not permit the manufacturer himself to extend the rating. A manufacturer desiring priority aid to secure materials for supplies sold under P-84 must still use the production requirements plan or the modified PRP.

HOW YOU CAN HELP CONSERVE ESSENTIAL METALS

"Users of valves and fittings can be of much assistance in several ways in helping conserve essential metals for the war effort. When bronze valves are replaced with new valves, turn the old valve over to a scrap dealer promptly. Specify iron valves in place of bronze valves wherever types are available and wherever it is practical to do so. Regrind regrinding valves, and in numerous other ways, salvage and save."—ALFRED J. EICHLER, vice-president, Walworth Co.

PD-1X TO SIMPLIFY DISTRIBUTOR PRIORITY PROBLEMS

Priority problems of distributors, wholesalers and jobbers will be simplified by the use of a new application form which has been designed for their special use, according to the WPB. The new form is known as PD-1X.

Insofar as materials and supplies can be made available without interfering with the war effort, priority assistance will be given to distributors, wholesalers, etc., who apply on the new form so that they can keep sufficient stocks on hand to maintain essential productive and service industries in operation.

In recent months, distributors have been hesitant to make deliveries to retailers, restaurants and other important users who cannot furnish priority rating certificates, because the distributors were afraid that they would not be able to replace the material in their own inventories. Use of the new form will enable distributors to request preference ratings for essential supplies without receiving or extending a rating on every individual order which they fill.

Distributors, wholesalers and jobbers who purchase the following supplies from producers will be entitled to apply for preference ratings on form PD-1X: automotive, aviation, builders, construction, electrical, foundry, hardware, health, industrial, heating and plumbing, railroad, refrigeration, restaurant, transmission, textile mill, welding and cutting.

LIMITATION ORDER CURTAILS USE OF FUEL OIL

In a move to conserve stocks of fuel oil in coastal areas, installation of new fuel oil burning equipment in 17 eastern states, the District of Columbia, Oregon and Washington was discouraged by limitation order L-56, issued March 14 by the director of industry operations of WPB and effective immediately.

The order forbids delivery of fuel oil for use in any new equipment unless installation is completed within 30 days, or in any converted facilities unless the conversion was completed within 10 days. An exception is made in the case of new construction if foundations are completed within 30 days and if fuel oil burning equipment is specified in the construction contract. Otherwise, no exceptions are to be made unless expressly authorized by the director of industry operations.

Consumers already using fuel oil were also forbidden to accept additional supplies unless they are making full use of standby facilities using fuels or power other than electricity or natural gas. Suppliers are not allowed to make deliveries of fuel oil to such consumers unless the standby facilities are being used to the fullest possible extent.

The order also provides that the director of industry operations may examine fuel oil burning facilities already installed, and suggest conversion for use of a less scarce fuel. After notice sufficient to permit such conversion, the director of industry operations may forbid further deliveries of fuel oil to the consumer who fails to convert.

The order provides for appeals in cases where its terms might work special hardship. The order does not apply to use of fuel oil in internal combustion engines.

STORAGE OF REFRIGERATION A METHOD OF INCREASING AIR CONDITIONING CAPACITY

The storage of refrigerating effect — accumulating "cold" by freezing ice during offpeak hours— offers to air conditioning engineers a method of overcoming priorities difficulties and increasing the capacity of an air conditioning system without increasing the peak electrical demand, says C. J. Otterholm, of McQuay, Inc. With a storage refrigeration job, a smaller compressor operating longer hours a day (usually 20 hours out of the 24) stores up cooling effect by freezing ice during the night and other offpeak hours. This cooling effect is used during the cooling peak in the afternoon by allowing the ice to melt. By proper control, water at a fixed temperature can be supplied the air conditioning coils at all times when it is needed.

On one job mentioned by Mr. Otterholm it was necessary to increase the capacity of a small air conditioning system without benefit of priority ratings. Spray nozzles were installed ahead of the direct expansion coil. Flooding nozzles were also installed to flood the coil, and flooding increased the coil's heat transfer coefficient by 15 to 20 per cent. Both sets of nozzles were supplied 37 F water from a storage or accumulator tank, and the net result was to increase the capacity of the existing system about one-third. The compressor is operated at night to freeze ice, and during the day to serve the direct expansion coil.

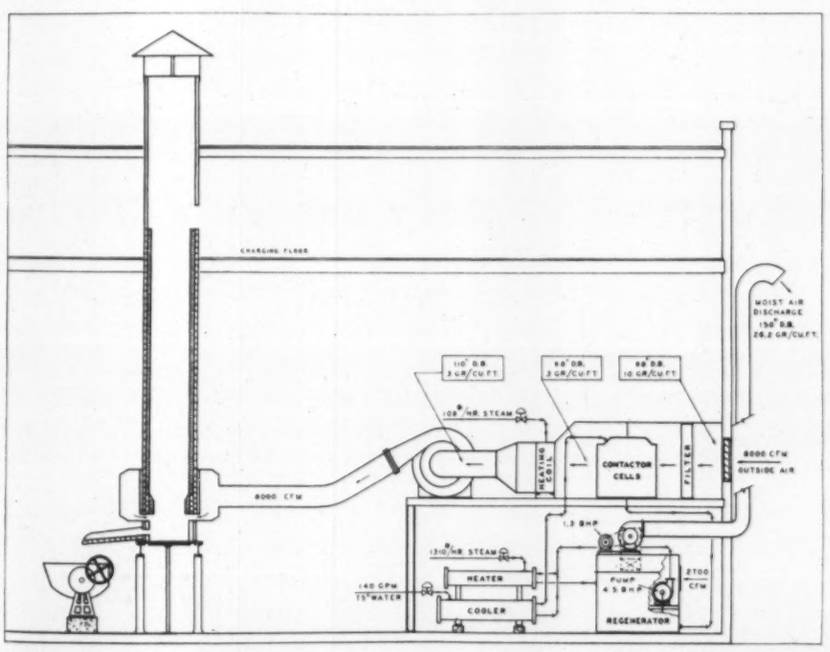
On another job where the building served is used only on occasion, it is desired to store refrigerating effect when there is no occupancy, and to use this cooling effect for air conditioning the building when it is occupied. The air conditioning system is put into operation a few hours before the building becomes occupied in order to pull down conditions to the proper point.

The consulting engineer specified that during this pull down period, sensible heat only should be removed from the circulated air; it would be wasteful to dehumidify the circulating air during the pull down period, as by the time the building became occupied, that particular air would be miles away. After the building is occupied, both sensible and latent heat must of course be removed.

This situation was met by arranging the control to hold a water temperature above the dew point of the air during the pull down period, lowering the water temperature after occupancy so that the air is both cooled and dehumidified. Water temperature on storage jobs is maintained at the desired point by varying the proportion of chilled water taken from the storage tank and the return water bypassed around the tank.

The economic application of storage refrigeration depends primarily upon the electrical rate structure and particularly the relationship of the peak demand charge and the cost of electricity during offpeak hours. Additional factors which should be considered these days are the difficulties of buying other equipment and refrigerants to increase capacity of an air conditioning plant, and *perhaps* possible restrictions in use of electricity in some areas, if this becomes necessary due to the demands of the production for victory program.

CHEMICAL MOISTURE CONTROL SYSTEM TO IMPROVE BLAST FURNACE OPERATION



Courtesy Surface Combustion.

A contract has been awarded for a lithium chloride moisture control system for furnishing dry blast to a new blast furnace of the Republic Steel Corp., according to F. M. Johnson, manager of the Kathabar division of Surface Combustion. The system is to treat 90,000 cfm of air and is to operate at one, two or three grains of air moisture, depending upon seasons. Such flexibility of operation provides for constant uniformity of product throughout the year.

The moisture control system will require only one 50 hp motor for power, and no materials used in the equipment are vital to war indus-

Schematic diagram of a chemical moisture control system for a foundry cupola. The principle is much the same for a blast furnace, except that the air requirement may run from 80,000 to 100,000 cfm instead of 8000 to 10,000

tries. It has been estimated that 10 per cent increase in furnace capacity, due to the application of the dry blast, would supply the necessary materials for the equipment in one day's time. Due to the simplicity of the system, the entire installation will be complete in less than 90 days' time.

Although this installation represents the first for blast furnace application, lithium chloride moisture control installations for cupola dry blast have been in operation since 1938. To date, there is a total of five cupola installations and two more are to be installed within the next few weeks.

LOW PRESSURE HEATING BOILERS SIMPLIFIED

Elimination of metal jackets, fusible plugs, and tri-cocks from low pressure heating boilers after June 1 was ordered by the War Production Board last month in schedule III to limitation order L-42. The order, however, grants permission to producers to deliver any boilers in stock June 1, even if equipped with the jackets, plugs and cocks, provided the boilers were completely finished by that date.

In addition, the regulations required that production of the metal jackets be ended on March 16. No limitation is placed on the manufacture of fusible plugs and tri-cocks because they are used for purposes other than low pressure boilers.

Producers of metal jackets are permitted to deliver jackets which were in stock in finished form on March 16, or processed so that the use of the metal for other purposes would be impractical.

Approximately 18,000 tons of steel are expected to be saved by elimination of the metal jackets, and 180,000 pounds of brass by elimination of the tri-cocks and fusible plugs.

Limitation order L-42 in general covers the simplification of heating and plumbing equipment. Schedules requiring the simplification of valves and pipe fittings were issued previously.

METHOD OF LIMITING ELECTRIC POWER USE

A procedure of curtailing use of electric power in order to make it available for war production purposes is illustrated by the WPB's recently issued limitation order L-46, which can be used for this purpose when shortages occur in Buffalo, Niagara Falls and western New York. The order was necessary, according to the WPB, because under certain combinations of lake level, air temperature, wind and ice, the supply of water in the Niagara River is reduced, resulting in a substantial though temporary reduction in electric power generation.

The order provides for:

1) Mandatory integration of power systems to develop the maximum use of power resources when needed to relieve shortages in the Niagara Falls frontier area.

2) Mandatory operation of generating equipment owned by industrial consumers to relieve the power shortage.

3) Mandatory reduction up to 50 per cent in the power demands of large power consumers (with demands over 200 kw) who are not engaged in war production or essential civilian services. This reduction is made during periods of power shortage only when the availability of power supply is insufficient to meet the demand after the steps mentioned above have first been taken.

4) Mandatory reduction on a pro rata basis in the power demands of large industrial users at Niagara Falls. This step is taken only after the first three above steps.

5) Mandatory restrictions upon the connection of new large industrial consumers with demands exceeding 100 kw, and upon expansion of loads of existing consumers.

Residential consumers are not affected.

AIR CONDITIONING TO INCREASE PRODUCTION OF SILVERY PIG IRON

Air conditioning, credited with increasing production in the iron and steel industry by as much as 27 per cent per blast furnace, will be utilized to increase production of silvery pig iron, vital to machine tool and other war production, it was announced recently by J. I. Lyle, president of the Carrier Corp. The announcement followed the signing of a contract with the Jackson Iron and Steel Co. for the air conditioning of one of the few high silicon content, or silvery pig, iron producing furnaces in the world.

Three furnaces have been air conditioned at the Woodward Iron Co., where production increases have been reported as high as 27 per cent per furnace with a saving of as high as 13 per cent in coke. Four installations are under way at Bethlehem Steel Co. and air conditioning equipment is in production for the Tata Iron and Steel Co., Ltd., in India, largest steel mill in the British empire.

Air conditioning equipment provides increased production of more uniform quality iron, according to Mr. Lyle, by removing as high as 30 tons of water per day from the 2700 tons of air blown through the average size blast furnace every 24 hr.

Describing the importance of such installations to the war effort H. R. Shick, general manager of Jackson Iron and Steel Co., said: "Silvery pig iron, containing 5 per cent to 17 per cent silicon in contrast to the low silicon content of basic iron, is used for special castings, many of which are in demand for the manufacture of machine tools. It is also used to help produce economical mixes for iron foundries and steel plants as a source of silicon, as a deoxidizing agent and to retard the oxidation of the other elements in the charge.

"In the production of silvery pig iron, coal as well as coke is used in charging the furnace.

"Air conditioning equipment to be furnished consists of a centrifu-

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AIR CONDITIONING GOES TO WAR

Air conditioning is one of the most important factors in America's war effort, William B. Henderson, executive vice-president of the Air Conditioning and Refrigerating Machinery Association, declared recently. Yesterday people thought of air conditioning as the acme of luxurious living, while today air conditioning is a front line fighter in our battle, he said. He predicted that history may well class industrial air conditioning among the important "weapons" with which America defeated the totalitarian powers.

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gal, turbine driven refrigeration machine of 300 ton cooling capacity, a spray type dehumidifier and a cooling tower.

"The air conditioning system used in the Jackson furnace will be 'pre-compression refrigerated dry blast,' under which the air is dehumidified by cold water sprays before delivery under pressure into the furnace." Because the air is cooled and thus contracts, a greater volume of air can be blown into the furnace at less cost for fuel. Often the savings in fuel used in the blowing engine alone are sufficient to cover the costs of operating the air conditioning equipment, Mr. Lyle states.

HEATING SECTION REQUIREMENTS OF NEW DEFENSE HOUSING CRITICAL LIST

As noted briefly in the March HPAC, a new Defense Housing Critical List has been issued by the housing priorities branch, division of industry operations, War Production Board. This list (effective February 24, 1942), is more specific than the original issued on September 19, 1941, (interpreted September 24, 1941) which it supersedes and nullifies. It is based upon the present critical position of many materials essential to the construction and equipment of housing, and is subject to revision as changes in the situation develop.

Preference ratings assigned to deliveries of scarce materials for defense housing projects may be applied *only* to items appearing on the new critical list, which was drawn up to conform to the limitations placed upon the uses of a number of scarce metals since the issuance of the original list last September.

The following are the paragraphs from the heating section of the new list. There are also sections on general provisions, structure, electrical, plumbing and gas distribution, household equipment, and land development and utility service:

500 HEATING

510 General:

511 The maximum net hourly output capacity of the heating unit or system as determined in 514 or 515 hereof is the capacity—after deductions have been made for piping and pickup, attached domestic water heaters, and non-dwelling heating loads—available to provide for the total hourly heat loss of the dwelling it heats. Such maximum net hourly output capacity in Btu shall not exceed 66 times the dwelling area in square feet or 80,000 Btu per dwelling unit, whichever is the smaller.

512 The total hourly heat loss of a dwelling shall not exceed such maximum net output capacity and shall be determined in accordance with the data and methods described in the current edition of the Guide of the American Society of Heating and Ventilating Engineers or by an alternate method which results in not less than the amount determined by the Guide method. Such total hourly heat loss shall be based on maintaining 70 F inside the dwelling when the outside temperature is at the design temperature for the locality. (Spaces such as unheated garages, attics and basementless spaces shall be calculated at outside design temperature.) Storm windows and doors furnished shall be credited with the heat loss reduction they effect.

513 Dwelling area is the total area used for dwelling purposes contained within the exterior walls at each principal floor level excluding garage and unfinished storage space but including only the finished area of any living, sleeping, dining or kitchen space located in the basement or attic.

514 Prior to 60 days after the effective date of this Defense Housing Critical List, the maximum net hourly output capacity shall be determined from catalog ratings in effect on said effective date. On or after 60 days after the effective date of this Defense Housing Critical List only equipment rated in accordance with 515 hereof shall be eligible. Equipment may be rated within a range of specified firing rates selected by the manufacturer if at any firing rate within that range its performance certified by the manufacturer to have been determined by test is within the limitation of the applicable code or standard specified in 515.

515 Except as provided in 514 hereof, the maximum net hourly output capacity is determined as follows:

5151 *Floor furnace:*

Gas fired—90 per cent of published, listed, or labeled output rating determined in accordance with the current Commercial Standard or Recommended Commercial Standard for Gas Floor Furnace—Gravity Circulating Type, or 90 per cent of the AGA output rating.

Oil fired—90 per cent of the manufacturer's certified output rating at 70 per cent or higher efficiency.

5152 *Space heater:*

Hand fired or coal or wood—90 per cent of manufacturer's specified output rating at 60 per cent or higher efficiency.

Gas fired—90 per cent of AGA output rating.

Oil fired—100 per cent of the published, listed, or labeled rating determined in accordance with the current Commercial Standard or Proposed Commercial Standard for Flue Connected Oil Burning Space Heater Equipped with Vaporizing Pot Type Burners.

5153 *Pipeless gravity furnace:*

Hand fired coal—90 per cent of the manufacturer's certified register output at 55 per cent or higher efficiency.

Oil fired—90 per cent of the manufacturer's certified register output at 70 per cent or higher efficiency.

5154 *Gravity furnace:*

Hand fired coal—110 per cent of the manufacturer's certified Standard Gravity Code rating in square inches leader area times 136 Btu.

Mechanically fired (conversion installations) same as for hand fired.

Gas fired (furnace-burner unit) 75 per cent of AGA bonnet output rating.

Oil fired (furnace-burner unit) 75 per cent of the manufacturer's certified bonnet output at 70 per cent or higher efficiency.

5155 *Forced warm air furnace:*

Hand or mechanically fired coal (gravity rated furnace with fan conversion) 100 per cent of the manufacturer's certified Standard Gravity Code leader pipe area in square inches times 180 Btu.

Gas fired (fan-burner-furnace unit) 85 per cent of AGA output rating.

Oil fired (fan-burner-furnace unit) 85 per cent of the output rating as determined by the Recommended Code of the NWAHACA for Testing and Rating Oil Fired, Fan-Furnace Combinations.

Stoker fired (fan-burner-furnace unit) 85 per cent of manufacturer's certified output rating at 70 per cent or greater bonnet efficiency.

5156 Boiler (only for heating systems serving two or more dwelling units or for extensions of existing plants to service additional living accommodations).

Gas fired—65 per cent of AGA output rating.

Hand fired or mechanically fired (all fuels) 100 per cent of current IBR net rating for cast iron boilers, or 100 per cent of net rating certified by the manufacturer to have been obtained by test procedure in accordance with the IBR Testing and Rating Code less (in each case) 12,000 net Btu per dwelling unit

for domestic hot water heated indirectly by the heating boiler.

516 Limitation on gas fired equipment: The use of gas fired equipment for heating space is subject to the availability of natural or mixed natural and manufactured gas from the utility company servicing the project. In addition, in areas where prohibitions or restrictions on deliveries of such gas for gas fired equipment for heating space are imposed by the War Production Board, proof must be submitted establishing exemption from such prohibitions or restrictions.

PERMISSIBLE INSTALLATIONS

520 Overflow heaters:

521 Floor furnaces and pipeless gravity furnaces—ferrous metal—no metallic coating.

5211 Registers—stamped, fabricated or cast ferrous metal—no metallic coating.

522 Space heaters, stoves and other similar heating devices—ferrous metal—no metallic coating.

530 Warm air distribution systems:

531 Furnaces—ferrous metal—no metallic coating.

5311 Fans, blowers and motors—for forced warm air systems.

5312 Filters—for forced warm air systems—nonmetallic containers and filter material with ferrous metal grid.

5313 Humidifiers — including parts and piping nonmetallic or ferrous metal—nonmetallic coated except for optional zinc coated piping.

532 *Distribution materials:*

5321 Ducts, connections, fittings, hangers, and fastenings—(*a*) ferrous sheet metal, untreated or phosphate treated (*b*) zinc coated ferrous sheet metal fittings for concealed composition duct work.

5322 Limitations—eligible materials and quantities limited to those necessary to meet the minimum sheet metal requirements of the current standards of the National Board of Fire Underwriters, Pamphlet No. 90. The following are not eligible: Sheet metal double ducts, sheet metal return ducts beyond 6 ft from the heater, sheet metal heavier than 26 U. S. ga.

5323 Registers and grilles — (*a*) stamped, fabricated or cast ferrous metal —no metallic coating.

540 Steam or hot water systems:

(only for heating systems serving two or more families or for extensions of existing plants to service additional living accommodations).

541 Boilers—ferrous metal with brass safety devices and cocks limited to minimum practicable and with pressure gages for steam boilers and combination temperature and altitude gages for hot water boilers.

Metal jackets not allowed.

542 *Distribution materials:*

5421 Piping—ferrous metal—no metallic coating.

5422 Fittings—cast ferrous metal—no metallic coating.

5423 Valves — no metallic coating — brass for 1½ in. size and less; ferrous metal elsewhere except seats, discs and stems; (*a*) flow control valve for forced circulation hot water systems; (*b*) check, blowoff, gate, globe, cock, radiator; (*c*) air valves.

5424 Traps, for multi-family steam systems (*a*) radiator — brass, no metallic coating; (*b*) boiler return, blast, bucket, float and thermostatic.

5425 Radiators and convectors — ferrous metal, no metallic coating and no metal enclosure.

5426 Hangers, brackets and fastenings —ferrous metal, no metallic coating.

543 *Accessories:*

5431 Pump assemblies—water circulators, condensation and return line vacuum.,

5432 Expansion tanks for hot water systems—ferrous metal, no metallic coating.

550 Firing equipment:

551 Oil and gas burners.

552 Coal stokers, hopper type — for multi-family systems only. No metallic coating.

553 Oil storage tanks—ferrous metal, no metallic coating, size not to exceed 275 gal capacity for tanks serving three units or less and not more than 100 gal per dwelling unit for tanks serving four dwelling units or more.

554 Oil line pumps.

555 Vent, fill and oil line piping and fittings—ferrous metal, no metallic coating.

560 Control equipment:

561 Electric material as allowed under 330, 340, 350, 360, and 370 of Electrical Section.

562 Relays and solenoids.

563 Damper regulators.

564 Thermostats (room, aquastats, airstats, combination fan and limit controls, combustion safety controls.)

565 Pressure controls.

5651 Pressure reducing valves.

5652 Pressure stats.

5653 Pressure relief valves.

566 Shutoff cocks for gas heaters, furnaces and boilers.

570 Breechings and smoke pipe.—Ferrous metal, no metallic coating. Breechings for multi-family heating systems may include necessary cleanout doors.

580 Vents and flues.—For heating equipment in demountable houses—ferrous metal, no metallic coating. Vents may include necessary casings, supports and connections. "Demountable houses" may include all prefabricated structural enclosures for dwelling purposes.

CONSERVATION OF MAN POWER TO INCREASE WAR PRODUCTION

The Department of Labor has inaugurated a program for conservation of manpower in United States war industries. Known as "CM", it is a war movement primarily aimed at increasing war production by preventing the work accidents that slow down the whole war machine.

Every accident in war factories is a setback to our whole fighting front and a delay to final victory. War workers can help the CM movement "by bowing to discipline like good soldiers and obeying every safety rule and edict laid down by the safety technicians in their plants. Accidents can't be killed—but they can be stopped."

Further information about this campaign can be had by writing the main office of the National Committee for the Conservation of Manpower in War Industries, United States Department of Labor, Washington, D. C.

High Pressure Valves—Formation of the high pressure valve industry advisory committee has been announced by the bureau of industry advisory committees, division of industry operations, WPB. Carl M. Lynge is government presiding officer, and W. Walter Timmis has been named alternate government presiding officer. Committee members are: L. D. Seymour, Pittsburgh Valve & Fittings Corp.; C. S. Bigelow, Milwaukee Valve Co.; and R. L. O'Brien, Detroit Brass & Malleable Works.

Pipe Covering—Ships are permitted to use high temperature pipe covering under conditions formerly forbidden, according to the terms of an amendment issued February 28 to the asbestos order, M-79. The amendment allows use of 85 per cent magnesia and other types on ship installations where temperatures under 200 F occur. Such insulations are restricted to temperatures over 200 F, except on ships.

Refrigerating Equipment Maintenance — Priority assistance for the maintenance of refrigerating equipment in stores and restaurants was granted by amendment No. 2 to preference rating order P-100, issued last month by the director of industry operations of the WPB. Under the terms of the order, a preference rating of A-10 is assigned to materials for repair and maintenance of equipment for specified industries and operations.

How Piping Is Welded for Navy Service

SUMMARY—The machinery installed on naval vessels has to be designed to meet a combination of service requirements which challenges the ingenuity of the builders; the primary concern is to build machinery which is reliable in operation, for a failure of even a minor part of the mechanical plant during a naval engagement might mean the difference between victory and defeat. . . . Welding applications in naval machinery were described at the annual meeting of the American Welding Society by Hugo W. Hiemke and John D. Bert, senior materials engineer and welding engineer (respectively) of the bureau of ships of the Navy Department; their paper was published in the Journal of the AWS, from which the following paragraphs on piping have been taken. . . . Summarizing their paper, the authors said that the advent of covered electrodes has revolutionized the fabrication of naval machinery. It has practically replaced riveting where that process was formerly used. Assemblies of simple steel castings, welded together, have replaced complex steel and gray iron castings. Many flanged and threaded joints in pipe lines have been replaced by welds. In another field, silver brazing has emerged as the predominant method of joining non-ferrous pipes. Hard surfaced disks and seats of steam valves have greatly simplified the maintenance of valves. . . . (Note: The statements made in this article are the opinions of the authors and are not the official views of the Navy Department.)

THE USE of welding for high pressure, high temperature steam lines for naval ships has been a gradual evolution from the van stoning and bending methods of fabrication. Both systems are being used in ship construction at the present time. In the future, it is likely that there will be a tendency for the elimination of all flanges but this change will be slow in coming due to the fact that there are many places where it is desired to have flanged connections so that sections of piping may be removed to provide space to service machinery.

Most of the welded high pressure steam lines are made of carbon molybdenum alloy steel and are welded in the shop using positioning equipment. The welded joints most used are shown in one of the sketches. Welding operators are qualified for welding high pressure piping by being required to make satisfactory welded pipe specimens in the horizontal fixed and vertical fixed positions.

When assembling piping and fittings that may require the use of a backing ring for welding it has been found necessary to expand tube ends to a predetermined inside diameter in order to line up the backing rings on the inside of the joint. The 15 per cent tolerance permitted on the thickness of piping produces unequal inside diameters of the piping and fittings.

Each welded joint in the high pressure piping system above 3½ in. in size is required to be radiographed in its entirety. Sizes 2½, 3 and 3½ in. are required to be inspected by means of partial radiography representing at least 60 deg of the circumferential seam. When defects are located by means of this partial radiography the entire circumference of such welds is radiographed, the necessary repairs effected, and radiographs of the repairs made. All radiographs are compared with the Navy's radiographic standards for class A-1 pressure vessel welds to determine the acceptability of the areas represented. Butt welded flange joints in the size range 2½ to 3½ in., inclusive, may be inspected by using magnaflux on the inside of the pipe in way of the weld after the backing ring has been removed.

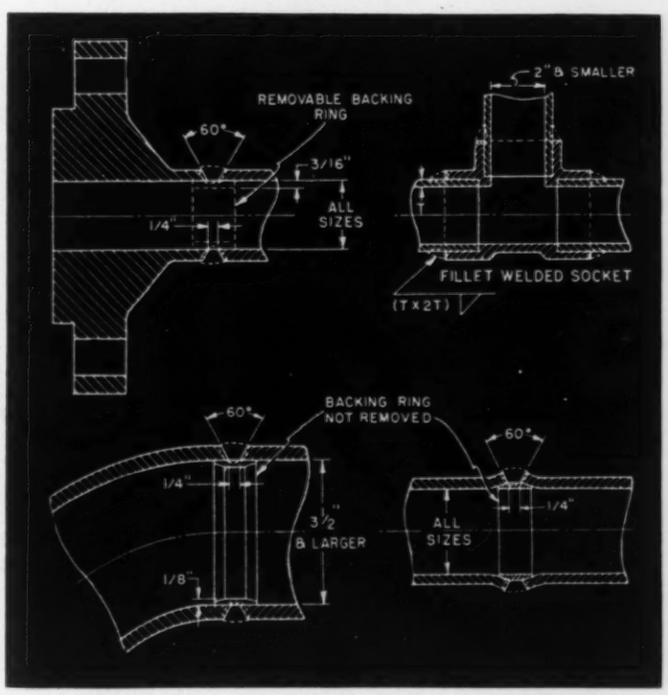
Disks and false seats for high pressure steam valves are faced with hard surfacing material. The part being hard faced is preheated before welding and allowed to cool slowly, usually by immersion in powdered lime or other heat insulating material. The inspection of these welded overlays consists of a careful surface inspection after grinding. For steam valve service, it is essential that these seats and disks be free of all defects.

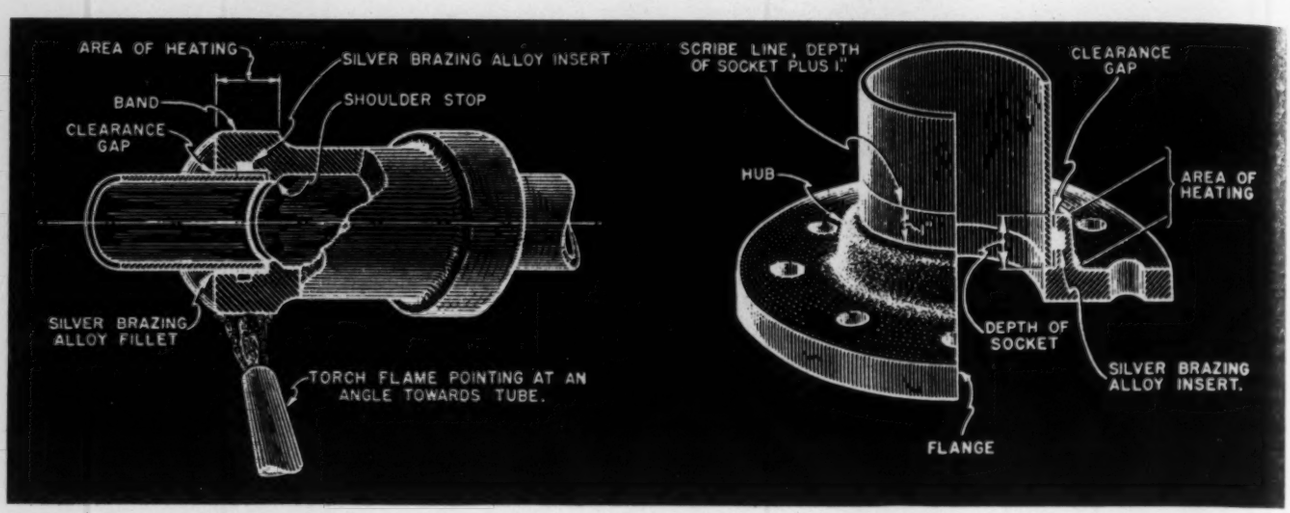
In addition to the high pressure steam piping a multitude of low pressure systems are entirely welded. The ferrous piping systems are either oxyacetylene or arc welded, and the non-ferrous systems are, for the most part, silver brazed.

The joint most used for brazing piping is the inserted ring fitting. This joint is used because it gives a visual indication of the completion of the joint, when a ring of silver brazing alloy forms at the mouth of the fitting.

Like all welding operations, silver brazing requires careful procedure control for its successful application. Operators must be qualified before proceeding with production brazing. Close tolerances between tubing and fitting are essential in order that the capillary forces may operate. This requires special tolerances on the outside diameter of the tube and close tolerances on the inside diameter of the fitting. The tube has to be trued up to a perfect circular

Typical welded joints used for welding high pressure steam piping for naval service



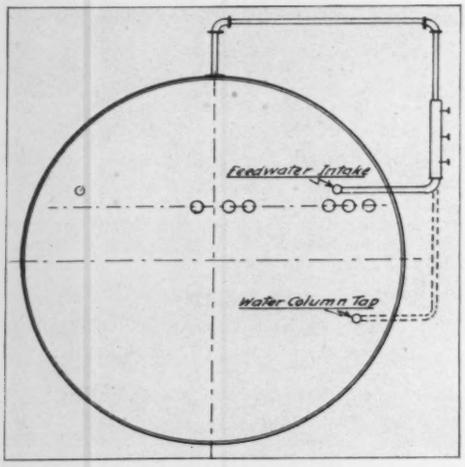


Inserted ring type fittings used for silver brazing piping aboard ship

shape before inserting it into the fitting. Proper cleaning and proper fluxing are essential.

Silver brazing is particularly advantageous for making joints in copper pipe lines, although it is also used on 70-30 copper nickel alloy, monel pipes and some steel pipes, such as lubricating oil pipes, which operate at relatively low temperatures. Appreciable weight savings have been effected by the replacement of threaded joints by silver brazed joints in pipe lines, and leakproof joints are readily obtained.

Change in Piping Corrects Trouble With Boiler Water Level



When difficulty was experienced in keeping the water line in sight on the gage glass, a check of the piping was made. When the lower end of the water column was connected as shown by the dotted lines the difficulty disappeared

WHEN TWO new low pressure, 60 in. horizontal tubular boilers were put into operation in a heating plant, the operators experienced considerable difficulty in keeping the water line in sight on the gage glass. Often—without apparent reason—the water would disappear completely, then gradually return to its level. The owner maintained that the steam outlet was too small and was causing a siphoning effect. The boiler manufacturer insisted that the trouble was in the feedwater and recommended the use of a boiler compound.

An investigation to determine the cause of the trouble disclosed that while the steam outlet was somewhat restricted, the velocity of steam flow was not excessive at the loads carried. Also there was a relatively small amount of raw water make up and the boiler water condition appeared to be satisfactory.

A check of the piping connections of the water column located the difficulty. The piping plans called for feeding return water through the blowoff valve connection. The piping contractor then connected the lower end of the water column into the tap provided for the feedwater line in its usual location, approximately at low water level, overlooking the tap left by the manufacturer for that purpose well below the water line. When the piping change was made the trouble disappeared.—R. S. HAWLEY, professor of mechanical engineering, University of Michigan; member of HPAC's board of consulting and contributing editors.

Vapor and Vacuum Specialties—Formation of the vapor and vacuum steam heating specialties subcommittee of the plumbing and heating industry advisory committee was announced March 4 by the bureau of industry advisory committees of the WPB. Jesse L. Maury is the government presiding officer, and the committee members are: John Bresnihan, president, Wm. S. Haines & Co.; William Jones, vice-president, Barnes and Jones, Inc.; B. F. Lerch, vice-president, Warren Webster & Co.; E. J. Ritchie, vice-president, Sarco Co., Inc.; Lewis Smith, C. A. Dunham Co.; and C. W. Stewart, assistant general sales manager, Hoffman Specialty Co., Inc.

"When we say 'Convert for war purposes', don't forget we mean convert for purposes of producing munitions of war and at one and the same time produce the essentials, the needs of our American economy, to keep it on a sound basis, so that we can produce more and more war material."—Donald M. Nelson, chairman, War Production Board.

Boiler Inspection—Firms engaged in examining or inspecting industrial installations for the purpose of discovering faults or defects were authorized to use an A-10 preference rating to obtain operating supplies by amendment No. 1 to preference order P-100 Functions of companies engaged in the inspection of boilers and other plant equipment are closely related to maintenance and repair, and the amendment gives such firms the benefit of the same priority rating which might be used by the companies for which they perform inspections.

AIR IN CONTACT WITH WATER

Analyzing Air Conditioning and Heat Exchange Processes with the Psychrometric Chart . . . By William Goodman

THERE IS hardly a process in air conditioning in which air and water are not brought into direct contact with each other. Because air conditioning equipment so frequently involves direct contact of water and air, a thorough understanding of the action that takes place between the water and air can be very helpful in analyzing the performance of equipment and in understanding the operation of air conditioning systems. The psychrometric chart lends itself beautifully to the analysis of the action of air and water in direct contact with each other.

No attempt will be made in this group of articles, of which this is the second, to analyze the performance of equipment. The action of air and water in direct contact with each other in different types of apparatus will be explained so that the differences involved in various types of conditioning processes can be readily understood.

The discussion is continued from the March issue:

Equilibrium Water Temperature in Nonadiabatic Straight Line Processes

There are also nonadiabatic processes for which the change in the state of the air takes place along a straight line—the ratio line. Although in an adiabatic process the slope of the ratio line depends only upon the initial temperature of the water, this is not true of nonadiabatic processes. In nonadiabatic processes the slope of the ratio line depends upon the amount of heat and moisture transferred between the air and water, and this, in turn, depends upon the area of heat transfer surface and the coefficients of heat and moisture transfer for the apparatus in which the process is taking place. In short, the slope of the ratio line for a nonadiabatic process and consequently the temperature that the water will assume —the equilibrium water temperature —depends upon the characteristics of the conditioning apparatus. As an example of a nonadiabatic process in which the change in the state of the air takes place along a straight line, see an earlier article on the evaporative condenser [HPAC, March to May, 1938]. The method of computing the equilibrium water temperature for the nonadiabatic process taking place in the evaporative condenser is also described there.

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Final Condition of Air

As is evident from the discussion in the preceding section, in nonadiabatic processes the slope of the ratio line and, consequently, the equilibrium water temperature, depends upon the characteristics of the conditioning equipment. On the other hand, in an adiabatic process the slope of the ratio line and, consequently, the equilibrium temperature, is independent of the characteristics of the conditioning apparatus. However, in any type of process, either adiabatic or nonadiabatic, the *final condition of the air* leaving the apparatus always depends upon such characteristics of the conditioning apparatus as the coefficients of heat and moisture transfer, and the areas of heat and moisture transfer surface. In addition, in an adiabatic process the final state of the air depends upon the weight of moisture supplied.

In general, the air will usually leave the apparatus in an unsaturated state, though it may be so nearly saturated that the difference cannot be measured. Theoretically, the air can leave the apparatus in a saturated condition only in an infinitely long apparatus. In any actual apparatus, point *4* of Fig. 1 [see p. 169, March, 1942, HPAC] can never coincide with point *3* although the two can be very close.

Change in State of Air in Contact with Water Whose Temperature Varies

Even though the change in the state of the air can be represented along a straight line only if the equilibrium temperature of the water remains constant, this principle can, nevertheless, also be applied to the analysis of problems in which the temperature of the water varies throughout the conditioning process. In this case the state of the air will change along a curve instead of a straight line. However, the straight line principle can be used to construct the curve along which the state of the air changes as the temperature of the water changes.

Whenever warm water is brought into contact with colder air, the water must cool as it surrenders heat to the air. If the weight of water supplied is so small that it evaporates completely, the water will cool to the equilibrium temperature. If the weight of water is slightly greater than the weight that can be evaporated, the water will cool almost to the equilibrium temperature. On the other hand, if the weight of water supplied is many times greater than the weight that can be evaporated, the water

SUMMARY—Processes in which air and water are brought into direct contact with each other are quite common in air conditioning; familiar examples are air washers, cooling towers and evaporative condensers. The condition of the air as it passes through such equipment can be represented either by a curve or straight line on the psychrometric chart. Such a curve is known as a "condition curve," because any point on the curve represents the condition of the air at various stages in the process. . . . . There is presented in the group of articles of which this is the second, the method of determining the state of the air while it is in contact with the water. Typical condition curves for a number of common processes are presented and discussed. A thorough understanding of the condition curves obtained for various types of apparatus will be of considerable value to those who design air conditioning systems. . . . . The author is consulting engineer, the Trane Co., and a member of HPAC's board of consulting & contributing editors

will not cool much; its final temperature will be close to the initial temperature. This will be discussed further in a later section entitled Limiting Final Temperature of Water.

For the present, it is sufficient to understand that the equilibrium and actual temperatures of the water are different only when the weight of water supplied is so small that all of the water is evaporated. On the other hand, when the weight of water supplied is many times the weight evaporated into the air, or condensed from it, the water will never cool to the equilibrium temperature. In this case, the temperature of the water with which the air is in contact at any instant will depend altogether upon the exchange of heat between the air and water as computed by means of a heat balance between the air and water.

If the temperature of the water changes, the change in the state of the air can no longer be represented by a straight line because the air at each point in its travel through the conditioning apparatus is brought into contact with water at a different temperature. Consequently, the mixing of the main stream of air with the air in the saturated surface film must take place along a series of straight lines, each of which has a different slope.

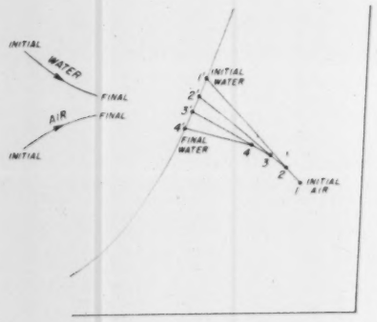


Fig. 4—Method of constructing condition curve for parallel flow of air and water when temperature of water varies

In a parallel flow spray type heating apparatus, the air and water flow in the same direction. In such an apparatus, the coolest air is first brought into contact with the warmest water, as shown in Fig. 4. Let point *1* represent the initial state of the air and point *1'* the state of the saturated air film at the initial water temperature. The initial change in the state of the air must take place along the line *1-1'*. However, as the state of the air changes to point *2*, the temperature of the water drops to point *2'*. Hence, the next mixing of air in the main stream with air in the saturated surface film on the water must take

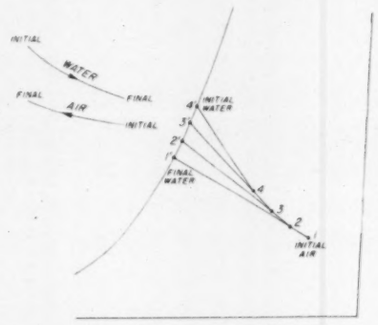


Fig. 5—Method of constructing condition curve for counterflow of air and water when temperature of water varies

place along the line *2-2'*. The state of the air now changes to point *3*, and the water temperature drops to *3'*. Points *1*, *2* and *3* are all very close to each other because the changes take place in an infinitesimal length of air travel and, consequently, only an infinitesimal amount of air from the surface film is mixed with the main air stream.

By repeating the process described in the preceding paragraph, and drawing a sufficient number of straight mixing lines in succession, the *condition curve*—the curve representing the condition or state of the air as it travels through the spray chamber—will be clearly outlined because it is the envelope of this series of straight lines. In Fig. 4, point *4* represents the final state of the air, and point *4'* the final water temperature.

For the actual work of drawing the condition curve by the method illustrated in Fig. 4, finite changes in the condition of the air must be used. However, by making these changes sufficiently small, the actual condition curve can be obtained with a good degree of accuracy.

For the actual work of drawing a condition curve, point *1* representing the initial state of the air and point *1'* representing the initial temperature of the water are first located on the psychrometric chart, and a straight line is drawn between them. Point *2* is then arbitrarily located close to point *1* on the straight line *1-1'*. The water temperature *2'* corresponding to point *2* can then be computed by means of a heat balance, or determined conveniently by means of the water-air chart described in HPAC, April to July, 1941.

A straight line is drawn between *2* and *2'*, and point *3* is then arbitrarily located close to point *2* on the straight line *2-2'*. The water temperature corresponding to point *3'* can now be computed in the same manner as *2'*, and the process repeated until the final water temperature corresponding to the final state of the air is reached. The smaller the distances selected between points *1* and *2*, *2* and *3*, etc., the more accurate will be the resulting condition curve.

In a counterflow heating apparatus, the flow of water and air are in opposite directions, as illustrated in Fig. 5. In counterflow, the warmest air is brought into contact with the warmest water, and the coolest air into contact with the coolest water. In Fig. 5, let point *1* represent the initial state of the air and point *1'* the *final* temperature of the water. A straight line is drawn between points *1* and *1'* because the cool entering air first comes in contact with the cool leaving water. Point *2* is arbitrarily located close to point *1* on line *1-1'*. The tem-

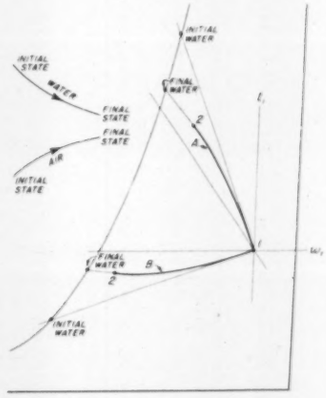


Fig. 6—For parallel flow, condition curves always bend *toward* the ratio line corresponding to the initial temperature of the water

perature of the spray water at each point can be computed by means of a heat balance equation or by means of the water-air chart. This process is repeated until the initial temperature of the water is reached and a sufficient number of straight lines have been drawn to clearly outline the condition curve.

For parallel flow of air and water in contact with each other, the

curves will be similar to those shown in Fig. 6 for both heating and cooling. The condition curves for parallel flow always bend toward the ratio line corresponding to the initial temperature of the water supplied, because (as will be shown later) the equilibrium temperature corresponding to the initial temperature of the water supplied is the lowest temperature to which water can be cooled or heated.

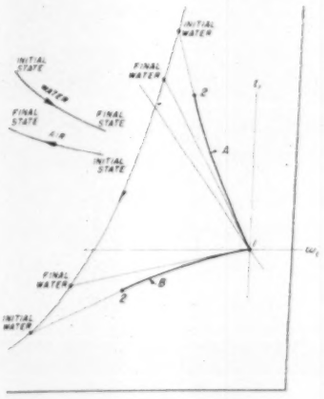


Fig. 7—For counterflow, condition curves always bend *away* from the ratio line corresponding to the initial temperature of the water

For counterflow of air and water in contact with each other, the condition curves always appear as in Fig. 7 for both heating and cooling. For this case, the curves bend away from the ratio line corresponding to the initial temperature of the water supplied.

When a coil is used to cool and dehumidify air, the water condensed from the air lies in a thin film on the metallic surface. The temperatures of the surface water and the surface film of air are the same, but the temperature of the cooling water inside the coil is lower than the temperature of these surface films. Furthermore, the temperatures of the surface films of water and air change progressively throughout the coil. The temperature of the surface films of water and air on the outside surface of a coil can be found by means of the water-air chart described in HPAC, April to July, 1941. Then by drawing a series of mixing lines between the state of the air and the state of the surface air film in exactly the same manner as described in the preceding paragraphs, condition curves for parallel and counterflow coils may be constructed.

Limiting Final Water Temperature

To some, there may be an apparent contradiction in the fact that in the earlier sections on equilibrium temperatures in adiabatic processes and nonadiabatic processes, the equilibrium water temperature—not the initial water temperature—is used, whereas in the preceding section the actual water temperature is used.

This apparent contradiction is due to the fact that the temperature of the water cannot change by more than a limited amount when the weight of water supplied is many times the weight evaporated. In this case, either all or the major portion of the latent heat required to evaporate some of the water is provided by the cooling of the water. On the other hand, when the weight supplied just equals the weight evaporated, even though the water cools to the equilibrium water temperature, it does not surrender sufficient heat for evaporating itself completely. The major portion of the heat must be supplied by the cooling of the air.

All of the foregoing can be visualized more clearly by using the equation below which is derived by writing the equations for the heat and moisture balances for the apparatus illustrated in Fig. 8, and then dividing the heat balance equation by the moisture balance equation.

Symbols

G = weight of dry air, lb per min;
G_t = weight of water supplied, lb per min;
h = enthalpy of air-vapor mixture, Btu per lb of dry air in the mixture;
h_{t1} = total heat of water supplied to conditioning apparatus, Btu per lb. of water;
h_{t2} = total heat of water leaving conditioning equipment, Btu per lb. of water;
q = slope of the ratio line on the psychrometric chart;
t_1 = initial temperature of water supplied to conditioning apparatus, F;
t' = wet bulb temperature, F;
w = absolute humidity of the air-vapor mixture, lb of moisture per lb of dry air in the mixture;
W = weight of water evaporated, lb per min.

$$q = \frac{G_t}{W}\left[h_{t1} - h_{t2}\right] + h_{t2} \quad \ldots\ldots [1]$$

When G_t/W, the ratio of the weight of water supplied to the weight evaporated, is very large, the temperature of the water drops from point 2 to point 3 of Fig. 9. The change in the state of the air

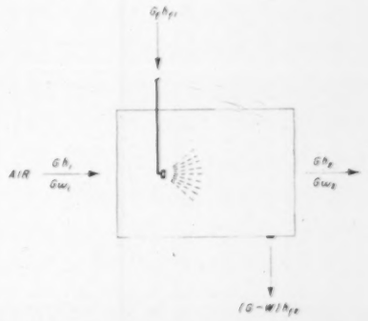


Fig. 8—Diagrammatic illustration of apparatus in which quantity of water supplied is greater than quantity evaporated

takes place along the condition curve through points 1 and 3. However, if the ratio G_t/W were reduced, the drop in the temperature of the water would be greater, say from point 2 down to point 4. In this case, the change in the state of the air would take place along the condition curve between points 1 and 4. If the ratio

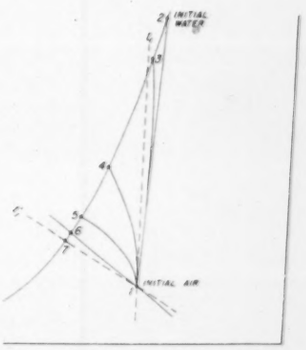


Fig. 9—Condition curves for various values of the ratio G_t/W

G_t/W were reduced still further and its value made very little larger than one, the temperature of the water would drop from point 2 to, say, point 5, and the change in the state of the air would take place along the condition curve between points 1 and 5. Finally, if the ratio G_t/W were made exactly equal to one, the temperature of the water would drop from point 2 to point 6. In this case, inasmuch as $G_t/W = 1$, equation 1 reduces to

$$q = h_{t1} = t_1 - 32 \quad \ldots\ldots [2]$$

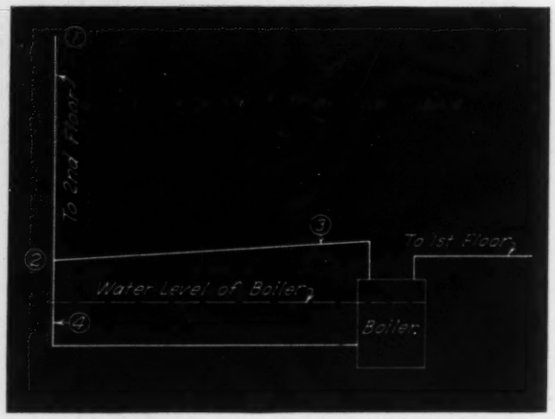
From [2], for which $G_t/W = 1$, the condition curve between points *1* and *6* becomes a straight line — the ratio line — whose slope corresponds to t_1, the initial temperature of the water supplied. Consequently, the lowest temperature to which the water supplied can be cooled is the equilibrium temperature as read at point *6*. This equilibrium water temperature can be obtained only when all of the water supplied is evaporated. The slope of the ratio line *1-6* corresponds, of course, to water at the initial temperature represented by point *2* of Fig. 9.

From the discussion in the preceding paragraphs, the following general principle is evidently true for either heating or cooling water with air: *When water is brought into actual contact with air, the final temperature of the water tends to approach the equilibrium temperature corresponding to the initial temperature of the water.*

Frequently, as is apparent from Fig. 2 [see p. 170, March, 1942, HPAC], the equilibrium water temperature is close to the initial wet bulb temperature of the air. Consequently, in actual computations it is quite common to assume that the initial wet bulb temperature of the air is the limit to which water can be cooled or heated when it is brought into contact with air. Though this assumption is not strictly true, it is sufficiently accurate for most practical work, and its use simplifies the computations of many practical problems.

[The third and last of these articles—of which this is the second—will appear in May.]

What's Wrong With This Heating System?



Sketch of one pipe steam heating system which is giving trouble. Mr. Lewis analyzes the difficulty and suggests the correction

REFERRING TO the accompanying sketch of a one pipe steam heating system, this was furnished by a correspondent who wishes aid in determining why a steam radiator at *1* on the second floor does not heat satisfactorily. The difficulty seems clear enough, even though we do not know the sizes of the pipes or radiators.

It appears that the branch main *3* running to the left of the sketch rises to a high point above the boiler and then pitches downward "several inches" in a distance which is not indicated to point *2* where there is a tee. From the top outlet of this tee the single pipe riser runs to the second floor, and from the bottom of the tee a pipe *4* continues downward to a wet return leading back to the boiler. The relation as to height of the tee at *2* to the boiler water level is the important matter, granting that the pipes are large enough and that they are not clogged.

The side inlet to the tee at *2* is only a trifle higher than the water level in the boiler or might even be level with or below the boiler water line. Then when the second floor is chilly, the rate of condensing steam in the second floor radiator increases rapidly, accompanied by a reduction in steam pressure. If the pressure within the pipe at *2* thus is reduced, steam will attempt to rush in through pipe *3*, and water also will try to equalize the pressure by rising out of the return *4*. If water thus seals the tee at *2* against entry of steam from pipe *3* the pressure in the second floor radiator will decrease rapidly (since its steam supply has been sealed off), and the pressure in pipe *1* will fall still further. Condensate will lie in pipe *3* and will cause water hammer complaint.

The cure lies in getting the crucially important tee at *2* higher above the boiler water level. If the tee at *2* is, say, 24 in. above the boiler water level it does not matter whether or not pipe *3* pitches up or down from the boiler to *2*. The size and length of pipe *3* may have much to do with the difficulty.—SAMUEL R. LEWIS, consulting mechanical engineer, and member of HPAC's board of consulting and contributing editors.

BIRKENSTOCK ELECTED HEAD OF NAFM

At the recent annual meeting of the National Association of Fan Manufacturers, held in Detroit, the following officers were elected for the ensuing year: J. M. Birkenstock, president; Edgar F. Wendt, vice-president; and L. O. Monroe, secretary-treasurer. At the same time the association celebrated its 25th anniversary.

UNIT HEATER MAKERS COOPERATE WITH WPB

At its annual meeting held on March 4, the Industrial Unit Heater Association elected E. W. Petersen president, A. B. Donkersley vice-president and L. O. Monroe, secretary-treasurer for the ensuing year.

The association has been cooperating with the War Production Board to standardize on the number of sizes of unit heaters and the substitution of materials to help conserve copper for the war effort.

GRAPHICAL ANALYSIS OF PERIODIC LOADS

By B. F. Raber and F. W. Hutchinson

Professor and Assistant Professor, respectively, of Mechanical Engineering, University of California

THERE IS real need for a simple, direct, non-technical method of determining, with accuracy, the effect on heating and cooling requirements imposed on a structure by periodic changes in the outside temperature. Solutions of the general transient problem have been proposed by various writers. A graphical solution adapted to the specific problem of daily periodic variation is here developed. This problem is important in the design and performance analysis of heating and air conditioning systems and, for such use, the advantages of this graphical system are:

1) Only one calculation is needed and that consists merely of substituting physical constants of the wall (obtainable from a handbook) in a simple algebraic equation.

2) The method consists of following a straightforward routine procedure which is exactly the same for all homogeneous walls under any form of inside and/or outside periodic temperature change.

3) The initial temperature gradient through the wall need not be known and regardless of how poor the guess may be, a correct result is unavoidable if directions are followed. Even if the heat flow was assumed to be from low temperature to high temperature the graphical construction would automatically compensate for the estimator's poor judgment and lead to a correct answer. However, accurate judgment is repaid with a more rapid solution.

4) A single solution for a particular wall is applicable to a wide range of temperature limits and, subject to one limitation, will serve for summer as well as winter load calculations.

5) One solution provides complete data on temperatures at *all* points in the wall for *all* hours of the day.

6) A solution for one particular wall is valid for all other walls of equal thickness and of corresponding physical characteristics.

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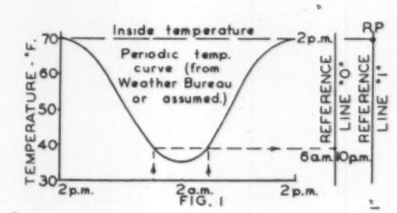

FIG. 1

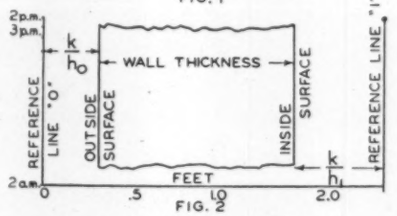

FIG. 2

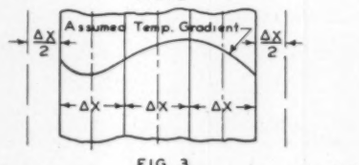

FIG. 3

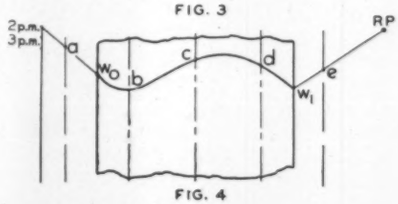

FIG. 4

Procedure

Fig. 1 — Select vertical temperature scale and horizontal time scale. Plot curve of daily outside temperature variation from weather bureau data, or approximately by drawing a sine curve through maximum and minimum temperatures. Draw in a vertical line labelled "Reference Line *O*" and transfer to this line points representing the temperature at each hour of the day. *Example:* From 6 a.m. on horizontal scale rise to intersect temperature curve, then move horizontally to right to intersection with reference line and mark this point 6 a.m.; note that the same point also represents 10 p.m. In this way establish points on the outside reference line *O* to represent the temperature at each hour of the day. Similarly draw an inside reference line *I* and transfer temperatures in the same way. (For the example, the inside temperature is held constant at 70 F so the inside reference line shrinks to a reference point, *RP*).

Fig. 2—To right of reference line *O* establish a horizontal scale in units of feet. Determine the conductivity, k, of the wall and the outside film coefficient, h_o [lacking other data h_o can be taken as $1.6 + 0.3v$ for smooth wall (ASHVE Guide) where v is the wind velocity in miles per hour.] Divide k by h_o and lay off a distance equal to the quotient in feet to the right of the reference line; this establishes the position of the outside surface of the wall. Lay off, in feet, a distance equal to wall thickness, thereby establishing position of inside surface of the wall. From inside surface lay off a distance k/h_i (h_i being the inside film coefficient calculated approximately from

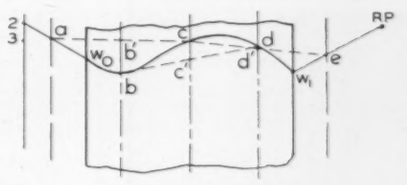

FIG. 5

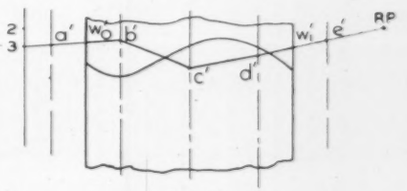

FIG. 6

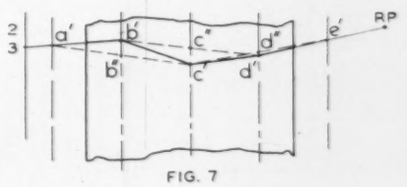

FIG. 7

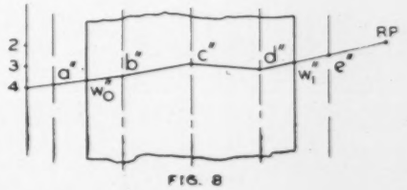

FIG. 8

the equation given above) to reference line *1* on which the location of *RP* is known. In this example a homogeneous wall is considered; this is not a required limitation.

Fig. 3—Determine the density, ρ, in pounds per cubic foot and the specific heat, c, of the wall. Select the interval of time, $\Delta\theta$ in hours for which the temperature change is desired and calculate the corresponding width of wall section, Δx, across which temperatures are to be determined; use the equation,

$$\Delta x = \sqrt{\frac{2k\,(\Delta\theta)}{c\rho}}$$

If the width of section calculated from the above equation is not equally divisible into the overall wall width, or if it is too large a percentage of the overall width (usually Δx should not exceed 33 per cent of the total) then a smaller $\Delta\theta$ is chosen and the calculation repeated. One operational difficulty of the graphical method for periodic problems is that the total period is 24 hours so selection of $\Delta\theta$ less than one-half hour leads to a large number of steps to complete the cycle.

Having determined Δx divide the wall into sections of width Δx. The analysis will give the temperature at the *center* of each such section so dashed lines are used to split sections and similar dashed vertical lines are located a distance $\Delta x/2$ out from the inside and outside surfaces. A temperature gradient through the wall at some arbitrarily selected starting time is estimated—or simply guessed.

Fig. 4—This figure represents the set up as determined in Fig. 3, but with all non-essential lines removed and important points of intersection lettered. The assumed temperature gradient through the wall at 2 p.m. is given by line $w_o bcdw_1$. Connect *RP* with w_1 to establish e and similarly fix a.

Fig. 5—To determine gradient at 3 p.m. connect a and c to fix b'. Connect b and d to fix c'. Connect c and e to fix d'.

Fig. 6—Connect *RP* and d' to fix w' and e'. Connect the outside reference point for 3 p.m. to b' to fix a' and w_o'. Connect $w_o'b'c'd'w_1'$ which would be the temperature gradient at 3 p.m. if the gradient at 2 p.m. were correct.

Fig. 7—To determine gradient at 4 p.m., repeat procedure of Fig. 5 connecting alternate primed points and thereby fixing double primed points, b'', c'', d''.

Fig. 8—Fix a'', w_o'', w_1'', e'' as in Fig. 6, but using primed instead of unprimed points and connecting to outside reference point for 4 p.m. Repeat the steps shown in Figs. 5, 6 and 7 until enough time intervals (24 for the example) have elapsed so that the temperature gradient for 2 p.m. is reached. If this agrees with assumed value, solution is realized. If it disagrees the construction is repeated for as many cycles as are needed to cause the gradient at a given time to take a fixed and unchanging position. In most cases superposition will be reached after one, or at most two, cycles.

Fig. 9—Note that the typical solution shown in this figure is valid for any wall of material for which $k/c\rho$ has the value for which this solution was obtained. Note the addition of a summer temperature scale. The solution is valid, summer and winter, for any temperatures or temperature ranges other than shown subject only to the limitation that the constant inside temperature must be equal to the maximum winter outside temperature or the minimum summer outside temperature. Other conditions can, of course, be treated similarly.

Fig. 10 — Surface temperatures are taken from the inside wall surface and plotted on a temperature-time field. From this curve the variation of load with time and data for calculation of maximum load can be determined.

TEMPERATURE GRADIENTS IN AN 8" CONCRETE WALL SUBJECT TO PERIODIC VARIATION OF OUTSIDE TEMPERATURE AND CONSTANT INSIDE TEMPERATURE

(Outside air temperature assumed to increase as a sine function of time for 12 hours, then decrease similarly for 12 hours. Inside air temperature assumed equal to maximum outside temperature (for winter conditions) or minimum outside temperature (for summer conditions)).

The solution given is valid for any temperature range subject to the limitations stated above.

Numbers represent half hour intervals from time at which minimum temperature (in summer) or maximum temperature (in winter) occurs. (Thus point 8 shows that assumed outside air temperature 4 hours after maximum outside temperature is equal to 68°F. for winter conditions.

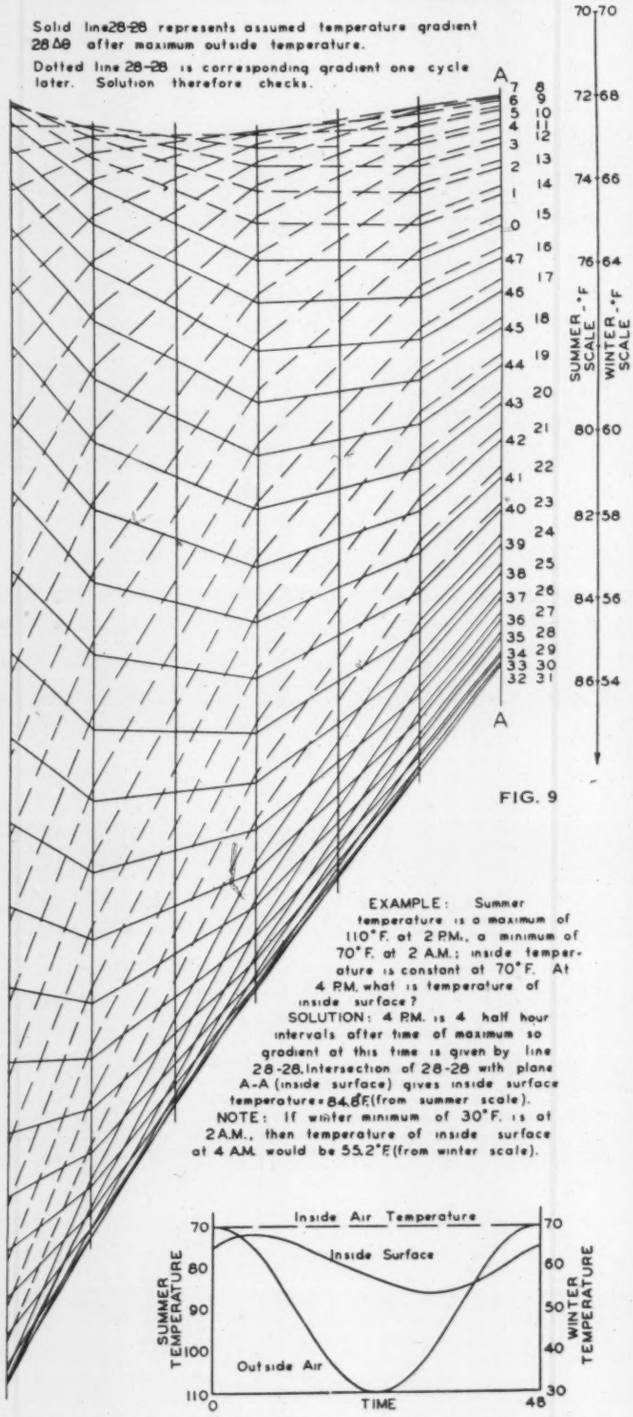
Solid line 28-28 represents assumed temperature gradient 28 Δθ after maximum outside temperature.
Dotted line 28-28 is corresponding gradient one cycle later. Solution therefore checks.
A
A
SUMMER SCALE °F
WINTER SCALE °F
70 70
72 68
74 66
76 64
80 60
82 58
84 56
86 54
EXAMPLE: Summer temperature is a maximum of 110°F. at 2 P.M., a minimum of 70°F. at 2 A.M.; inside temperature is constant at 70°F. At 4 P.M. what is temperature of inside surface?
SOLUTION: 4 P.M. is 4 half hour intervals after time of maximum so gradient at this time is given by line 28-28. Intersection of 28-28 with plane A-A (inside surface) gives inside surface temperature = 84.6°F (from summer scale).
NOTE: If winter minimum of 30°F. is at 2 A.M., then temperature of inside surface at 4 A.M. would be 55.2°F (from winter scale).
Inside Air Temperature
Inside Surface
Outside Air
SUMMER TEMPERATURE
WINTER TEMPERATURE
TIME
0
48

FIG. 9

FIG. 10

Heating · Piping and Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

Attention!

The regular quarterly meeting of the Council has been called by Pres. E. O. Eastwood for April 9 at the Engineers Club of Philadelphia. All members of the Council have been invited by Pres. H. B. Hedges to attend the monthly meeting of Philadelphia Chapter on that evening when President Eastwood will be the speaker.

• • •

The number of new members elected during the first quarter is impressive and the Membership Committee invites the active assistance of every member throughout the year.

• • •

Engineers, especially those with heating and ventilating experience, are in demand for war plants and by the Army and Navy. There is a place for every engineer, either in civilian ranks or in the armed forces, to make a maximum contribution to victory.

• • •

The Society has offered all of its research facilities to the government of the United States and President Eastwood has appointed a War Service Committee to be of service to any governmental agency.

• • •

Enlist with the Engineers and/or Buy a War Bond Today.

CONTENTS

APRIL, 1942

The Effect of Attic Fan Operation on the Cooling of a Structure, by W. A. Hinton and A. F. Poor. 244

The ASHVE in the War, by S. H. Downs..... 251

Observed Performance of Some Experimental Chimneys, by Richard S. Dill, Paul R. Achenbach and Jesse T. Duck..... 252

Saint Paul in June 261

Proceedings of Annual Meeting 1942..... 262

Local Chapter Reports..... 273

Candidates for Membership 275

Candidates Elected 276

Officers, Council and Committees..... 277

To All Research Technical Advisory Committees and Society Members

It is the desire of the Committee on Research to lend all possible aid to the Government in this Emergency. For this reason, the schedule of work at our Laboratory has been only tentatively arranged, in order that provision may be made for emergency investigations as the need arises.

Most of the Technical Advisory Committees, which met in Philadelphia, were asked to give special consideration to the War in determining their program. However, only two have, as yet, produced fitting proposals.

All Technical Advisory Committees and all other Society members are invited—are urged—to submit suggestions for study by our Laboratory, or by our Co-operating Institutions, in order that the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS may best serve the American Nations.

F. C. McIntosh, Chairman
COMMITTEE ON RESEARCH

The Effect of Attic Fan Operation on the Cooling of a Structure

By W. A. Hinton* and A. F. Poor**, Atlanta, Ga.

This paper is the result of research sponsored jointly by the Atlanta Chapter and the Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the State Engineering Experiment Station at the Georgia School of Technology.

SUMMARY—The cooling effects of an attic fan were observed under actual use conditions. A differential of 2 F between inside and outside temperature was obtained with 45 air changes per hour. An identical house without an attic fan was found to be as much as 8 F warmer. Air changes exceeding 45 per hour were found to be uneconomical

Introduction

DURING THE summer months the minimum outside air temperature at night is considerably lower than the maximum temperature of the preceding day. Data from the U. S. Weather Bureau indicate that in Atlanta, Ga., this temperature difference averages approximately 18 F. The outside temperature usually starts to drop in the afternoon and the decrease is quite rapid during the early evening hours. By 10 p.m. approximately two-thirds of the temperature drop has occurred.

A house and its contents can be cooled during the late afternoon and early evening by drawing through it the cooler outside air. This permits the walls, ceilings and floors to transfer heat from both inside and outside surfaces, reducing the amount of heat that must be conducted through the building materials to the outside surface. The flow of the cool air increases comfort conditions in the house during the cooling period.

In recent years attic fans have been installed in a number of houses in this section of the country. These fans are mostly of the axial or propeller type and they are usually installed so that outside air is drawn in through the house and discharged from the attic. Various arrangements have been used in making the installations.

With the increasing use of these fans a need was felt for further study of their performance. Through the cooperation of the Atlanta Chapter and the Committee on Research of the ASHVE, this investigation was undertaken to determine methods of fan selection, operation and installation which would bring about a maximum cooling of the structure and thereby afford the greatest comfort to the occupants consistent with economy of installation and operation. It was also desired to obtain data by which cooling effects of the fan could be compared with the natural cooling of a house. This paper describes the method used to obtain this information and contains preliminary data obtained from test runs made during the summer of 1941.

A considerable amount of information has been compiled by Badgett on the installation and use of attic fans.[1] Much of this information is of a general nature regarding structural details and methods of operation. Some cost data were also included.

The use of night air for cooling was investigated by Kratz, and Konzo.[2] The night air was used to reduce the daytime cooling load on a cooling system in the Research Residence at the University of Illinois. Outside air was drawn through the house at night while during the daytime the house was kept closed. With this arrangement requirements for artificial cooling were decreased.

Further information on summer cooling studies carried on at the Research Residence has been reported also by Kratz and his co-workers.[3]

Test Equipment and Its Arrangement

In order to determine more accurately the cooling effect of an attic fan it was decided that temperature measurements should be made in two identical houses located next door to each other. Two such houses were found and a fan was installed in one of them. Thermocouples were installed in each of the rooms of both houses. The house without a fan, which will be referred to as the control house, was used to determine the cooling effects caused by natural circulation. The added cooling effects of the fan in the test house could, therefore, be calculated.

The Houses

The two houses selected are single story frame structures possessing identical floor plans. They are the same size and are constructed of the same kind of material. The houses rest on piers and the floors are approximately 18 in. above the ground. The distance between houses is approximately 12 ft. Neither house has a basement. Details of construction are tabulated in Table 1.

Each house has a volume of 8,900 cu ft. Volumes of individual rooms and the areas of floors, windows and door openings are tabulated in Table 2. The windows and doors are not weather stripped and neither house has awnings over the windows.

The floor plans of the two houses are shown in Fig. 1. Rooms in corresponding positions in the two

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[1]The Installation and Use of Attic Fans, by W. H. Badgett. (*Bulletin*, Agricultural and Mechanical College of Texas.)
[2]Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo. (ASHVE TRANSACTIONS, Vol. 40, 1934.)
[3]Study of Summer Cooling in the Research Residence for the Summer of 1934, by A. P. Kratz, S. Konzo and M. K. Fahnestock. (ASHVE TRANSACTIONS, Vol. 41, 1935.)

Table 1—Structural and Thermal Properties of the Houses

	Type of Construction	Coefficient of Heat Transmission *U*
Floor	Pine on pine with building paper between	0.34
Outside Wall	2 in. x 4 in. Studs on 16 in. centers inside ½ in. sheet rock, outside ⅜ in. lap siding on ⅞ in. pine sheathing	0.25
Roof	Composition shingles on ¾ in. pine	.....
Ceiling	½ in. sheet rock. 2 in. x 6 in. joists on 16 in. centers	0.437[a]

[a] Combined roof and ceiling coefficient.

Table 2—Dimensions of Rooms and Openings in the Houses

Room	Room No.	Volume Cu Ft	Floor Area Sq Ft	Windows[a]	Doors[b]
Living room	I	1935	193.5	2	2
Front bed room	IV	1950	195.0	2	2
Rear bed room	V	1420	142.0	2	2
Kitchen	II	2200	220.0	2	2
Hall	III	1390	139.0	0	2
Total	.....	8895	889.5	8	10

[a] Wood sash double hung 33x72 in.
[b] Wood panel 3 x 7 ft.

houses were used for similar purposes. Fig. 1 also shows the location of the thermocouples and the location of the intake grille of the fan in the test house.

Both houses were occupied during the tests. The test house was occupied by two adults. The control house was occupied by three adults.

The Fan

The fan installed in the test house is an axial type, 36 in. in diameter and fitted with four blades. It is rated to deliver 10,000 cfm at a speed of 350 rpm and 0.01 in water gage static pressure. The arrangement of the fan installation is shown in Fig. 2. The opening in the ceiling was covered with an expanded metal grille having 70 per cent free area. The grille opening is 5 ft long and 3 ft 10 in. wide. Two ceiling joists run length wise of the opening, reducing the gross area to 17.5 sq ft.

With the fan installed in this manner the outside air could be drawn in through the windows and then through the rooms and the hall and finally through the grille opening. The fan discharged the air into the open attic space. A louvred opening in the gable supplemented by openings placed in the eaves enabled the air to escape from the attic to the outside.

An anemometer, range 200 to 2000 fpm, was used to measure the rate of air flow. This instrument was calibrated prior to its use in the tests. The calibration was made for this particular use. The fan was set up in the laboratory and equipped with the same intake box and grille that was to be used in the test house. The fan intake box and duct are shown in Fig. 3. A duct with pitot tube stations was attached to the discharge side of the fan. By placing different size orifices on the discharge end of the duct, varying rates of air flow were obtained. With the fan in operation, pitot tube readings were taken according to the Standard Test Code Procedure of the *National Association of Fan Manufacturers*. At the same time anemometer readings were taken under the grille at 15 stations. The average values of the air velocities at the grille were plotted against corresponding volumes in cubic feet per minute computed from the pitot tube readings. The curve obtained (Fig. 4) was used in computing the rates of air flow from anemometer readings taken during the test runs.

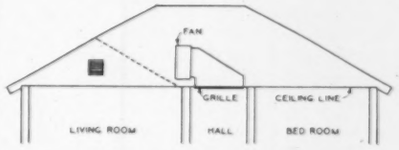


Fig. 2—Fan installation in test house attic

The Thermocouples

In the two houses 90 thermocouples (iron constantan) were installed. They were checked for accuracy by comparison with a mercury in glass thermometer which had been calibrated by the U. S. Bureau of Standards. Thermocouples were placed in each room and in the hall of the test house in the following positions: Breathing level; floor level; and ceiling. Horizontal distribution was used at each level. In addition, thermocouples were placed in contact with the walls and ceilings of each room so that surface temperatures could be measured. In the control house thermocouples were located at various levels in all rooms and in the hall. A few thermocouples were also placed in the attic, on the roof and just outside of both houses. The thermocouple locations are shown in Fig. 1.

The leads from all thermocouples were run to an instrument booth built between the two houses. They were connected through switches to a potentiometer. Fig. 5 shows the

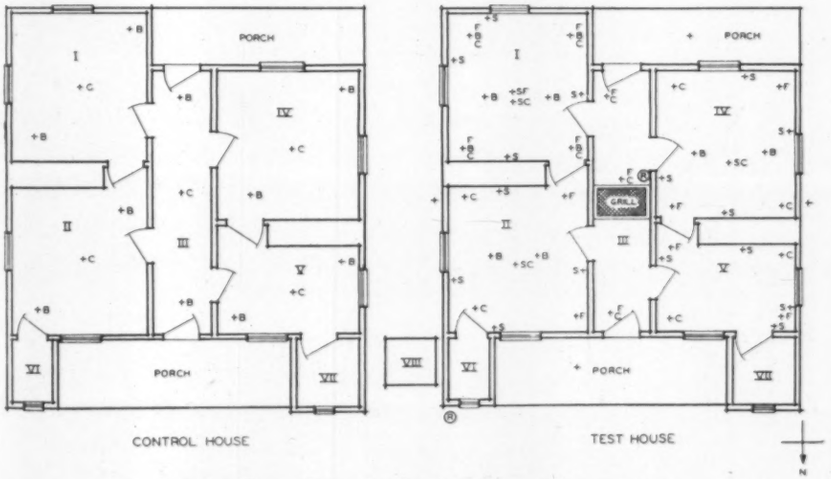


Fig. 1—Floor plans showing location of instruments

Key: I Living Room; II Kitchen; III Hall; IV Front Bed Room; V Rear Bed Room; VI Pantry; VII Bath Room; VIII Instrument Booth; R Location of Temperature Recorder Bulbs. Thermocouple locations are indicated by marks followed by letters indicating levels of positions, as follows: F Floor Level; B Breathing Level; C Ceiling; S Surface Temperature; SC Surface of Ceiling; SF Surface of Floor.

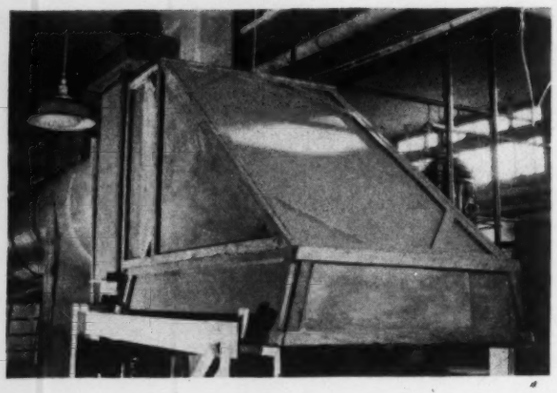


Fig. 3—The fan and duct as set up in the laboratory

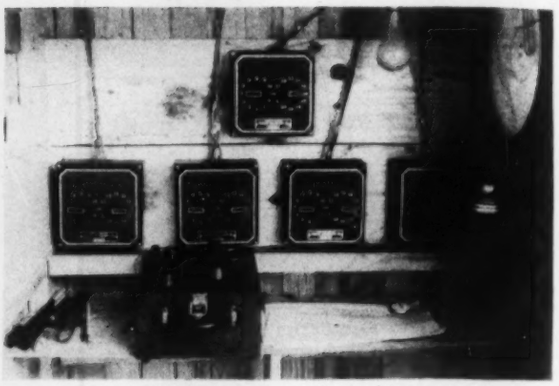


Fig. 5 — The instrument booth showing switches and potentiometer

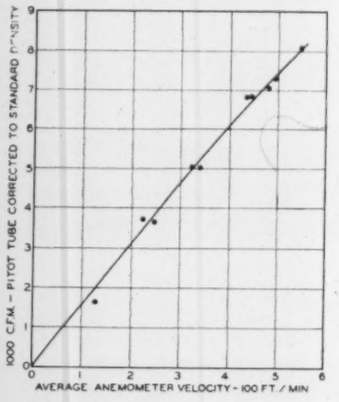


Fig. 4—Calibration curve of anemometer

arrangement of switches and potentiometer in the instrument booth.

In order to supplement the thermocouple readings, two temperature recorders were installed. These recorders were kept in adjustment to give readings the same as the calibrated thermometer. One recorder was set up in the hall of the test house. A record of the average inside air temperature could thereby be obtained. The other temperature recorder was installed in the instrument booth with the bulb located to give outdoor air temperatures.

Operating Procedure

Each test run involving the operation of the attic fan was carried out according to the following general outline. Just before the fan was turned on a set of thermocouple readings was taken. A set of readings consisted of one reading for each of the 90 thermocouples installed in both houses. The temperatures were recorded to the nearest ½ F. The windows and doors were opened or closed as desired and then the fan was turned on. Reading of the thermocouples was continued and complete sets of observations were made at approximately 30-min intervals during the run. The rate of air circulation for each run was determined by taking 15 anemometer readings under the grille and then using the anemometer calibration curve.

A number of variations were made on the above procedure. The time of starting the fan was varied from 12 noon to 5 p.m. During some runs the fan was turned off at approximately 9 p.m., while on others it was permitted to operate until early the next morning. The front and rear entrance doors were always closed during a run. On some of the runs one or more of the rooms was completely closed off. In such cases the volume of the open rooms through which the air was circulated was used to compute the number of air changes per hour. The amount of infiltration of air drawn through the closed rooms can be considered as negligible compared with the total volume of air handled by the fan.

One 24-hour run was made to compare the test house and the con-

Table 3—Tabulation of Data Taken on Typical Run

TEST HOUSE

Time of Day		3:25	4:05	5:00	5:30	6:30	7:30	7:55	8:20	8:55	9:25	9:55
Living Room Temp.	Floor	86.5	85.9	86.8	86.8	86.6	85.0	84.8	84.6	84.5	83.5	83.5
	B. L.	89.3	90.2	91.0	88.9	87.5	85.5	85.7	85.5	84.5	84.5	84.0
	Ceil.	92.0	91.3	93.0	87.0	90.0	87.5	86.5	86.5	85.0	85.0	84.0
Front Bed Room	Floor	87.5	91.2	89.5	87.8	85.5	82.8	82.5	81.8	81.2	80.0	79.0
	B. L.	89.0	91.8	90.5	89.0	85.5	83.5	82.5	81.2	81.0	79.5	78.5
	Ceil.	90.0	92.0	91.0	90.5	86.8	83.8	82.5	81.8	81.5	80.2	79.5
Rear Bed Room	Floor	85.8	89.2	88.5	87.8	85.5	82.8	82.0	81.8	81.0	80.0	79.2
	B. L.	87.0	91.5	89.5	87.8	86.5	84.0	83.2	82.0	81.5	80.5	79.5
	Ceil.	87.8	92.0	90.5	89.8	87.2	83.2	83.0	82.0	81.2	80.5	79.5
Kitchen Temp.	Floor	86.0	89.0	89.0	88.0	85.0	84.2	84.5	84.5	84.5	83.5	82.5
	B. L.	88.5	90.5	90.0	88.2	85.5	85.5	85.5	84.5	84.5	84.0	82.5
	Ceil.	90.8	91.8	91.5	90.2	87.0	85.2	85.5	84.5	84.5	83.5	82.5
Hall Temp.	Floor	85.8	89.0	87.8	87.5	84.7	83.2	82.2	81.8	81.0	79.8	79.7
	B.L.					None						
	Ceil.	87.3	91.2	89.5	89.0	86.2	84.7	83.0	82.5	81.3	80.7	80.5
Avg. Temp. Under Floor		83.0	84.5	84.0	84.2	80.8	80.5	80.0	79.8	80.0	78.8	78.5
Avg. Outside Temperature		91.0	91.2	88.0	86.2	82.2	81.2	81.5	80.2	79.8	78.2	76.0
Avg. Attic Temp		106.2	108.5	94.2	92.4	88.3	85.7	84.4	83.5	82.4	81.3	80.6

SURFACE TEMPERATURE

Living Room		89.1	90.3	89.7	88.8	87.6	85.8	85.2	85.1	84.4	83.4	82.7
Front Bed Room		90.0	91.3	90.6	89.3	87.3	85.3	84.2	83.2	82.7	81.6	81.1
Back Bed Room		88.6	91.1	90.4	90.2	87.7	85.2	84.0	83.2	82.8	81.6	81.0
Kitchen		89.1	90.3	89.5	88.4	86.5	85.5	85.2	84.2	84.0	82.7	82.3

CONTROL HOUSE

Living Room	B. L.	92.0	89.2	92.5	93.0	....	90.5	89.0	87.2	87.5	86.8	87.5
	Ceil.	95.0	93.5	94.5	94.0	....	92.0	89.5	87.5	82.5	87.0	89.0
Fr. Bed Room	B. L.	91.0	91.8	92.2	93.0	....	91.0	90.2	88.2	88.2	87.8	87.8
	Ceil.	95.0	95.0	96.0	95.0	....	92.5	92.0	91.0	90.0	89.5	89.0
Back Bed Room	B. L.	88.5	91.5	91.0	91.8	....	90.5	89.5	88.5	88.2	87.2	87.5
	Ceil.	93.5	95.0	95.0	95.0	....	92.5	92.0	91.0	90.0	89.0	89.0
Kitchen Temp.	B. L.	93.8	89.2	90.0	92.5	....	90.5	89.5	88.2	88.8	87.8	87.5
	Ceil.	97.0	94.0	94.0	95.0	....	92.5	91.5	91.5	90.0	89.5	88.5
Hall Temp.	B. L.	88.5	90.5	88.8	92.8	....	90.8	89.5	86.5	87.8	87.5	87.8
	Ceil.	88.0	92.0	92.0	94.0	....	92.5	92.0	88.0	90.0	89.0	89.0
Avg. Attic Temp		108.5	108.7	109.0	107.2	....	98.8	96.8	95.5	94.0	91.5	90.2

B. L. = Breathing Level.
Fan in operation from 4:50 to end of run.
Dark at 7:20.
Living room and kitchen in test house closed off.
Air changes per hour, 80.
All temperatures are averages of several thermocouple readings, degrees Fahrenheit.
Date of run, September 2, 1941.

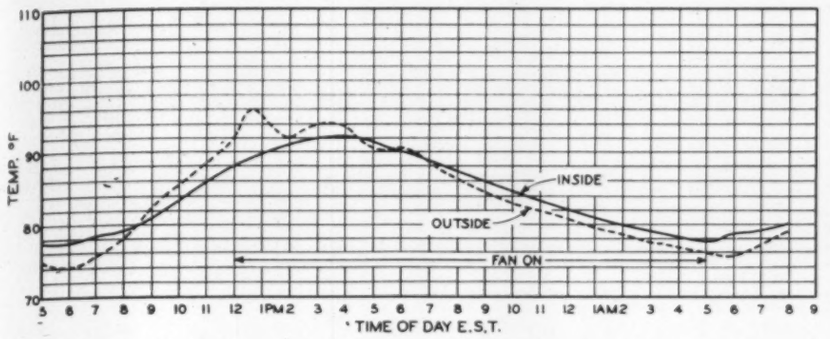


Fig. 6—Relation of time and average inside and outside temperatures (55 air changes per hour; July 30-31, 1941; weather, partly cloudy)

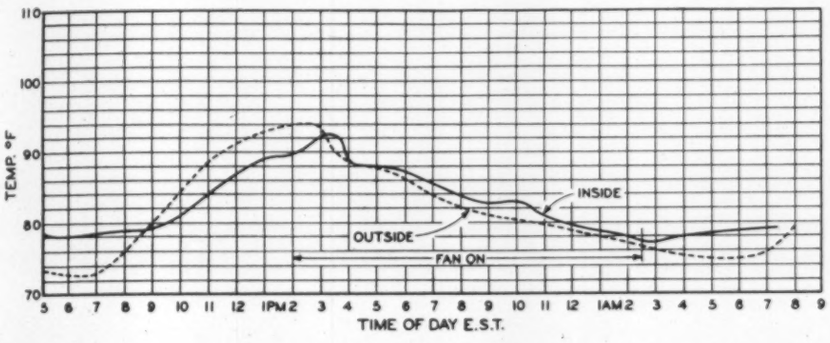


Fig. 7—Relation of time and average inside and outside temperatures (57 air changes per hour; August 9-10, 1941; weather, clear)

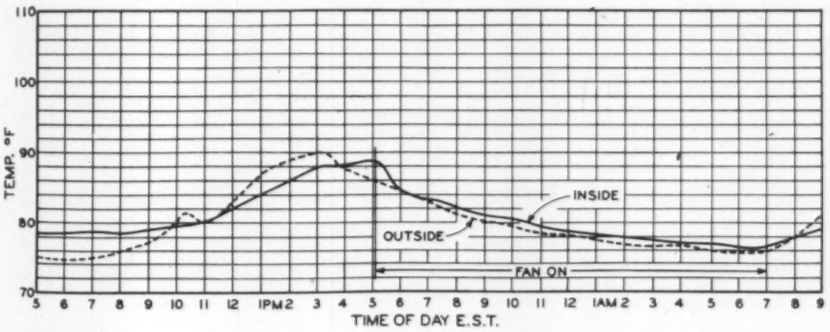


Fig. 8—Relation of time and average inside and outside temperatures (80 air changes per hour; September 3-4, 1941; weather, clear)

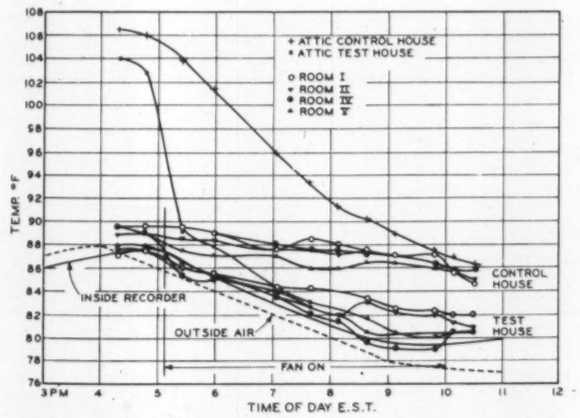


Fig. 9—Relation of time and individual room temperatures in test and control houses (60 air changes per hour; September 9-10, 1941; weather, partly cloudy)

trol house from the standpoint of their heat capacity, measured under normal air circulation conditions. In this run the grille opening to the fan and all the openings from the attic of the test house were closed off. The fan was not turned off. The door and window openings were identical in both houses. Thermocouple readings were taken every hour during the 24-hour period.

The personnel available did not permit the reading of the thermocouples to be continuous over 24-hour periods except during the one run mentioned above. The recording thermometers provided 24-hour records of the test house and outdoor temperatures. Since all the air drawn through the rooms of the test house entered the hall and was then drawn up through the fan, the temperature indicated and recorded by the thermometer placed in the hall was an average value of the temperatures of the rooms. A comparison of the various open room thermocouple readings with the readings of the temperature recorder in the hall shows that they are in close agreement (Fig. 11). Weather observations were recorded qualitatively.

Results and Discussion

During July, August and September, 30 test runs were made. The results from typical runs are presented in the tables and figures. The period of fan operation; the number of air changes per hour; and the temperature data are indicated on each curve. The temperature indicated for a particular room is the average of the readings of all breathing level thermocouples in that room. The average inside air temperatures of the test house, which

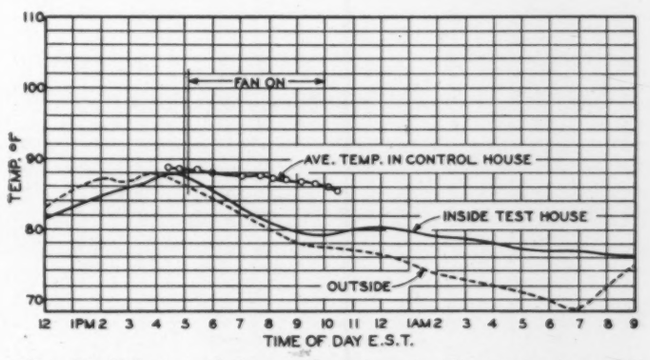


Fig. 10—Relation of time and average inside temperatures of test and control houses (60 air changes per hour; September 9-10, 1941; weather, partly cloudy)

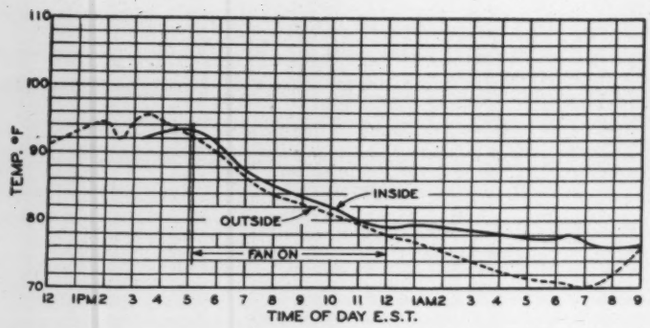


Fig. 11—Relation of time and average inside and outside temperature (80 air changes per hour; September 9-10, 1941; weather, clear)

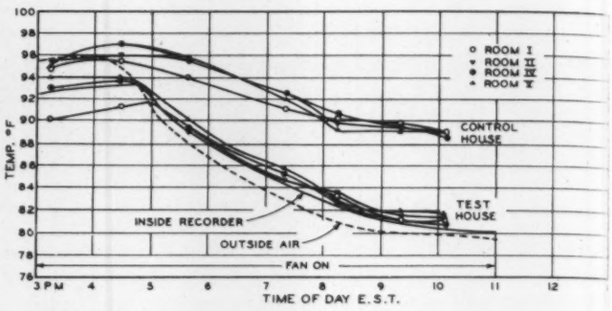


Fig. 12—Relation of time and individual room temperatures in test and control houses (46 air changes per hour; August 11-12, 1941; weather, clear)

are plotted on a number of the curves, were taken from the hall recorder charts.

The nature of the data recorded during each of the test runs is indicated in Table 3, which presents in tabulated form the various temperatures observed in both houses during a typical run. Attention is called to the fact that during this run the living room and kitchen windows were closed at 7 p.m.

All of the data obtained from the test runs have not been tabulated in this paper. Instead, most of the data presented are in the form of curves. Presentation of the results in this form simplified the analysis and the interpretation of the data.

During the runs indicated by Figs. 6 and 7 the living room of the test house was closed. The windows and doors of all the other rooms were open. In the run shown by Fig. 8 the living room and the kitchen were closed off.

Comparison of Figs. 6, 7, and 8 shows the effects of starting the fan at different times of the day. Fig. 8 shows that the outside temperature had reached its maximum and had started to decrease before the fan was turned on. It will be noted that within a few minutes after the fan was turned on the inside air temperature started to drop rapidly.

From Figs. 6, 7, and 8 it can be seen that the slope of the inside temperature curve approaches the slope of the outside temperature curve during the period of fan operation. With the fan in operation, the slopes of the two curves become practically equal within one to two hours after the outside temperature drops down to the inside temperature. When the slopes of the two curves are equal the rate of decrease in inside temperature is the same as that of the outside temperature.

From these curves it can be concluded that the fan should be turned on just as soon as the outside temperature drops down to the temperature on the inside of the house. During the summer months in Atlanta this time ranges from 4 to 5 p.m. It will be noted from all the curves presented that within two hours after starting the fan (provided the outside air temperature is dropping) the inside temperatures drop to within two or three degrees of the outside temperatures. After three to four hours of fan operation this temperature differential is less than two degrees.

Fig. 9 shows among other things that within one hour after turning on the fan during late afternoon the test house temperatures start dropping much more rapidly than the control house temperatures. It can further be seen that even five hours later the control house temperatures have dropped on an average of only three degrees. During this same time interval the outside air temperature has dropped approximately 10 deg and the average test house temperature has been lowered approximately eight degrees.

Figs. 8, 9, and 11 represent runs in which the fan was started at 5 p.m. and stopped at 7 a.m. the next day, 10 p.m. and 12 midnight, respectively. The inside and outside temperature curves for the test house indicate that within an hour after the fan is turned off the inside temperatures do not continue to drop at the same rate as the outside temperatures. Instead, the inside temperatures actually rise slightly (up to 1 deg) during this time interval and then start dropping again but at a much lower rate. This slight temperature rise is caused by the transfer of heat from the warmer walls and ceiling of the structure to the substantially still air in the house.

From the data presented it can be concluded that a one story frame structure, of low heat capacity, can be brought down to within a few degrees of the existing outside air temperature if the fan is operated for only three or four hours and that further cooling will occur, although at a lower rate, after the fan is turned off. The optimum time for turning off the fan would depend on the actual outside temperatures and on the individual personal preferences. In a better insulated structure the further cooling characteristics might be of a different nature. If the maximum lowering of the temperature of the inside air and of the structure itself is desired, the fan should be operated until approximately 6 a.m. the following morning. By following this procedure of all night fan operation the house would remain cooler a longer period of time during the following day, especially if the doors and windows were closed.

The effects of varying the number of air changes per hour on the temperatures inside the test house can be observed from Figs. 6, 7, 9, 11 and 12. A minimum of 45 air changes per hour (calculated) were used during this entire investigation. A larger number of air changes per hour were accomplished by closing off some of the rooms in the test house. The curves indicate that the differential reached between inside and outside temperatures, after a given number of hours of fan operation, was not measurably affected by increasing the rate of air changes above 45 per hour. The inside temperatures referred to here are for

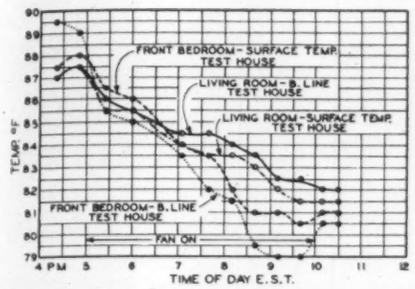


Fig. 13—Relation of surface temperatures to breathing level temperature (60 air changes per hour; September 10, 1941; weather, partly cloudy)

the rooms which were open.

In the run shown by Fig. 9, room I was closed at 7:00 p.m. while room II was closed at 8:15 p.m. From these curves it is seen that these two rooms do not cool as rapidly as the ones left open. The closing of rooms I and II increased the number of air changes in the rooms left open. Fig. 9 shows that the effect of increasing the rate of air flow did not produce any significant change in the rate of cooling of the rooms that were left open.

These results indicate that the use of air changes in excess of 45 per hour would be of little benefit so far as further cooling of a typical one story frame structure is concerned and would, therefore, be uneconomical. Further investigation may show that even less than 45 air changes per hour will be sufficient.

A comparison of the temperatures in corresponding rooms of the test house and the control house points out several advantages of the fan operation. Referring to Figs. 9 and 10, it can be seen that, with the fan in operation, the test house temperatures start decreasing just after the outside temperature drops below existing inside air temperatures, while there is a lag of one to two hours before the control house temperatures begin to drop. From Fig. 9 it can be seen that after several hours of fan operation, the individual rooms in the test house are kept five to eight degrees cooler than the corresponding rooms in the control house. It can also be noted that the temperature in the test house attic drops down very close to the temperature of the rooms soon after the fan is turned on, while the attic of the control house cools over a much longer period of time. The removal of this body of hot air from the attic space will obviously make the space below easier to cool.

Fig. 13 shows a comparison of average breathing level and surface temperatures for two rooms of the test house maintained under different conditions. With the fan in operation, the living room was kept closed while the other rooms were open. The surface temperatures in the living room were found to be slightly lower than breathing level temperatures. This indicates that some heat was being transferred from the air to the walls and ceiling. In the bed room, which was open and through which air was drawn, the surface temperatures were found to be higher than the inside air temperature, indicating that heat was being transferred from the surfaces to the air. The rate of decrease of surface temperatures was found to be very close to the rate of decrease in breathing level temperatures, as can be seen from Fig. 13.

A 24-hour test run was made to compare the temperatures of the test house and the control house when the fan was idle. The results of this run are shown in Fig. 14 in which the average breathing level temperatures are plotted for each house. These temperatures were observed hourly by taking thermocouple readings.

From the curves it can be seen that the temperatures in the control house were approximately two degrees higher than the test house temperatures, except during the cooking periods, when the temperature difference was greater, and during the period from midnight until 5 a.m., when the temperature difference was approximately one degree.

The higher control house temperatures can probably be attributed to the fact that a kerosene stove was used for cooking purposes while an electric stove was used in the test house.

Attic Openings for Egress of Air

At the beginning of this investigation, the louvred opening in the gable of the test house provided only two square feet of flow area for the exit of air from the attic. The resistance of air to flow through this small opening reduced the fan delivery. Other openings were placed in the eaves of the house to make a total of eighteen square feet. This increase in area of opening resulted in an increase in fan delivery from 6000 cfm to 7000 cfm.

After this an additional opening was placed in the attic and no appreciable increase in fan delivery was obtained. Since the gross area of the inlet grille was 17.5 sq ft, it can be concluded that the total area of the openings from the attic should be approximately equal to the gross area of the inlet grille.

Summary

An investigation has been made on the operation of an attic fan in a single story frame structure and the temperatures produced have been compared with those obtained at the same time from an identical structure not equipped with a fan.

The following conclusions can be drawn from the results obtained during thirty test runs, conducted during the summer of 1941, in which operating conditions and time intervals were varied:

1. When outside air is drawn through a typical single story frame structure at a rate of 45 air changes per hour, the inside air temperatures are lowered to within two degrees of the existing outside air temperature.

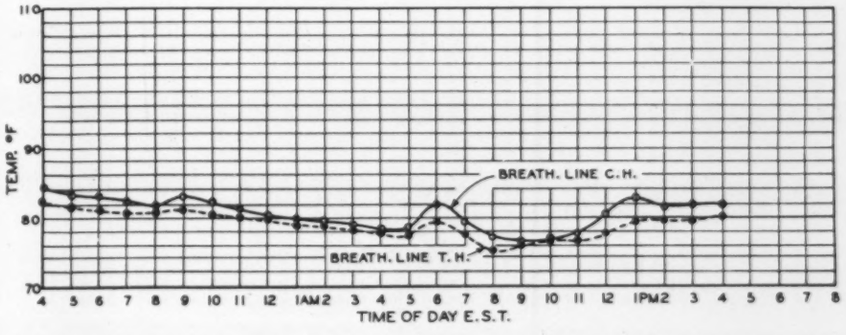


Fig. 14—Relation of temperatures in test and control houses with fan idle (September 24-25, 1941; weather, cloudy, precipitation 0.36 in.)

2. With 45 air changes per hour the differential between inside and outside temperatures is maintained effectively constant. The slopes of the curves for inside and outside temperatures are practically equal.

3. An increase in the number of air changes above 45 per hour does not bring about any appreciable reduction of this temperature differential, nor does it increase the slope of the indoor temperature curve.

4. The optimum time in Atlanta, Ga., for starting the fan was found to range from 4 to 5 p.m.

5. After the fan has been in operation for three or four hours in the evening it can be turned off provided the inside temperature has been lowered sufficiently unless continued operation is desired by the occupants to produce precooling for the following day.

6. The test house was found to be five to eight degrees cooler than the control house.

7. The total area of the openings in the attic for egress of air should not be less than the gross area of the intake grille.

Outline of Future Work

1. Investigation of effect of lowering number of air changes per hour.

2. A further study to establish effective temperatures in both the test house and the control house. (Some preliminary data, not included in this paper, were obtained during the present investigation.)

3. A study of attic fan installations discharging air directly to the outside, with attic as an inlet plenum.

4. A study of attic fan installations which will exhaust the house only at night and the attic only during the day.

5. A study of the resistance to air flow in typical attic fan installations including a comparison of installed delivery with free delivery.

6. Investigation of the combined effect of insulation and attic fan operation.

7. Investigation of attic fan operation in multi-story houses and in residences of different types of construction.

Acknowledgments

The authors wish to express their appreciation of the cooperation of J. A. Miller, president and general manager of the Exposition Cotton Mills, Atlanta, Ga., who made possible the use of the two houses. The cooperation of Mr. and Mrs. J. A. Towns and Mr. and Mrs. A. D. Garner, occupants of the houses, is appreciated also. The authors wish to thank Dr. F. C. Houghten; Prof. R. S. King; Prof. R. E. Lewis; and K. H. Henley, of the Georgia School of Technology, for their assistance rendered during this investigation. The authors wish to express their appreciation of the splendid cooperation of the Committee on Research of the Atlanta Chapter of ASHVE, H. King McCain, T. T. Tucker, C. Boynton Cole, F. Warren Clare. The authors also appreciate the help given by other members of the Atlanta Chapter.

DISCUSSION

W. H. Badgett, College Station, Tex. (Written): The authors have, I believe, overlooked one of the most important effects secured from the operation of an attic fan, and seemingly have thought of it largely only as a means of reducing the indoor air temperature, and that of the mass of the building. One of the greatest benefits resulting from the use of attic fans is in the early evening hours through introduction into the house of large volumes of cooler outdoor air in which the occupants may place themselves, thus enjoying the combined effects of lowered air temperature and increased evaporation of body moisture with a resulting lower effective body temperature. I believe that it is important to have a larger number of air changes in the early part of the night with possible reduced air movement for sometime after the occupants have retired in order to secure continued pre-cooling of the house for the following day.

In conclusion No. 7, the authors state that the *gross area* of the discharge openings should not be smaller than the intake grille. What they should have referred to is the *effective areas* of these openings, for *gross areas* can be almost anything, whereas the *effective area* is the controlling factor. Tests at College Station have indicated that more unsatisfactory operation will result from undersized discharge openings than from undersized intake grilles. These tests indicate, too, that where possible it is desirable that the net free area or effective area of the discharge opening be somewhat larger than the inlet. This avoids the possibility of building up a back-pressure with the resulting decrease in operating efficiency and increased noise level.

In the 24-hour test run to compare the two houses without the fan in operation, it is not indicated whether the intake grille of the fan was closed during this test. If the grille remained open, there was in all probability a natural air movement from the rooms of the test house out through the attic which would possibly account for the slightly lower temperature in the test house.

It is possible that both the intake and discharge openings in this installation are small for optimum results. However, since the authors have measured the air movement and converted it to air changes, this will not affect the findings in this paper.

Although there is little that is new presented in this paper, it is a valuable addition to knowledge in the field of attic fan operation because it verifies opinions and facts long held by those who have worked with attic fans. As I understand it, this is the foundation for extensive research in the operation of attic fans, and I believe the authors have made a splendid beginning.

In their outline of future work, I suggest that the following be included in their studies:

1. Resistance to air flow of various types of grilles, particularly expanded metal, wooden egg-crate, automatic intake louvers, automatic exhaust swinging doors, and screen wire when used to cover exhaust openings.

2. Determine if possible the over-all resistance of an average attic fan installation under normal operating conditions.

3. Determine the optimum grille velocity from the standpoint of efficient operation and satisfactory noise level.

J. H. Walker, Detroit, Mich. (Written): The research project reported in this paper is the first to be sponsored jointly by a chapter and by the Committee on Research, and opens up new possibilities in chapter activities. The Atlanta Chapter is certainly to be commended for its initiative and perseverance in conceiving and carrying through this project. There are undoubtedly other lines of research in which chapters might participate because there are many phases of heating and air conditioning which differ according to locality and which, therefore, must be studied locally. This paper shows an enormous amount of data considering the scanty funds which were available. The investigators apparently did a careful job.

If the project can be continued during the summer of 1942 there are two points that deserve attention. The first has to do with air velocities, which apparently have not been taken into account. As in previous discussions of the attic fan, too much significance has been attributed to the cooling effect on the structure and too little importance attached to the creation of air currents at proper locations in the rooms. The purpose of the attic fan is to make people comfortable, not to cool the structure, and to neglect air movement is to neglect one of the basic factors in comfort. The air velocity in the rooms is greatly influenced by the amount of opening of the different windows, and the cooling effect on the human body is correspondingly affected. Future work should study that factor.

The second point concerns the matter of mean radiant temperatures. No measurement of thermal environment is complete without including mean radiant temperature preferably measured with the

Vernon globe thermometer. Radiation from walls and ceilings is an important factor in attic fan operation. A comparison of the mean radiant temperatures in the test house and in the control house would be interesting and valuable.

J. A. BISHOP, Dallas, Tex.: The statement was made that people who lived in this house seemed to feel that they were more comfortable with the fan operating even at times when the results of this test indicated that the house was getting warmer, because of the introduction of higher outside air temperature. This situation parallels exactly the experience we have found in Texas. After considerable experience with attic fans it seems to be the general opinion that attic fans are started somewhere from 10 or 11 o'clock in the morning and that is contrary to the opinion of many.

I understand that some of the thermocouples were located on the ceiling of the room. When the outside temperature is warmer than the inside of the house, it is still possible that the attic temperature might be higher than the outside air. Under such conditions the ceiling separating the attic space and the occupied space would be somewhere between the outside and attic temperature. Therefore, the ceiling of the house would act as a panel heater and the use of an attic fan to remove the excess heat from the attic would thereby reduce the surface temperature of the ceiling and reduce the radiation. Have you any figures to substantiate this idea?

MR. HINTON: I do not have figures with me so it is impossible for me to give you a definite answer at this time. However, we had about the same thought that you have expressed to the effect that the operation of the fan during the day prevented an increase in attic temperature and reduced the ceiling radiation effect.

E. K. CAMPBELL, Kansas City, Mo.: This idea of air circulation for reducing air temperature was utilized some years ago on a church where the inside volume was much greater. In this case the heat capacity of the material was probably less in proportion and we found that while the fan was operating at night, drawing in air from the outside and using from 10 to 15 air changes per hour, we obtained fairly satisfactory results in cooling the interior of the building by morning. By opening the fan room door so as to utilize the air from the basement and closing the building tight we could hold the temperature to a fairly low value throughout the morning service of the church.

THE ASHVE IN THE WAR

By S. H. Downs, Kalamazoo, Mich.

IN THE present period of readjustment from peace to all out War, it is incumbent upon each member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS to analyze carefully the position of this International, scientific Society in the National effort to win the War; and thus preserve for posterity our democratic civilization, which we believe to be the best so far developed by the human race.

It is especially true that this is the responsibility of each member because the Society itself is a democratic type of organization, whose policies are determined by the trend of the desires of the members, within the constitutional limitations freely agreed to by all members; and because all members joined of their own free will and may leave at any time they see fit.

The decision, on the direction our Society activities are to follow, takes on much added significance when we scan what has already been done and what we are now doing.

In its essence, our problem is to decide whether our work has been so narrow in scope and viewpoint that we must change to some different activities, (and so change the whole order of the life of the Society), or whether our work has, and does now, include all of those activities which are essential to the sound prosecution of the war and to the welfare of the Nation and its people.

The Society as a whole directs its efforts to research, education and scientific development through the following media:

1. Holding meetings and discussions.
2. Publication of THE GUIDE, the JOURNAL, and the TRANSACTIONS.
3. Translation of research results to practical applications.
4. Development of Engineering Standards and Practices.

In the By-Laws of the Society, Art. B-1, *Sec. 1*, we find all of these activities defined as necessary to the accomplishment of the objectives of the Society.

These activities form a coordinated whole, whose parts are very definitely interrelated. As an instance, we see a steady flow of research data into the JOURNAL, the TRANSACTIONS, and THE GUIDE, and a continual expectancy on the part of the profession and the public that the Society will formulate standards and codes to put to use the knowledge gained through* "research, discussion, intercourse and publication".

The complete register of activities is implemented, and put into action, through the holding of meetings nationally, by chapters, and by committees.

The things so accomplished are manifold, and their importance is enhanced because they are widely distributed through publication and are used to an extent far beyond the relatively small membership of the Society.

They are used, directly and indirectly, by every industry in the country and by all the armed forces.

We do not heat, we do not ventilate and we do not air condition merely for the sake of creature comforts.

We do these things to make possible, the year around, activities that otherwise would be dormant or greatly handicapped during large portions of the year.

Without these things, the war effort of this country would be a puny thing.

With these things, the United States will soon have the greatest offensive and defensive power on earth.

The present emergency calls for an expansion of effort on the part of the Society to the end that more research data and more knowledge be made available for use.

Every member can help by being more active in national and chap-

[Concluded on p. 260]

Presented at the Engineers Club, Omaha, Nebr., March 11, 1942.

*Quotation marks indicate direct quotations from the Constitution and By-Laws.

Observed Performance of Some Experimental Chimneys

By Richard S. Dill*, Paul R. Achenbach** and Jesse T. Duck†, Washington, D. C.

SUMMARY—Two small experimental chimneys arranged to operate at various stack heights were tested at several rates of gas flow and inlet temperature to simulate residential performance. Temperature gradients throughout the height of each chimney are reported, together with draft conditions and friction losses.

Introduction

PUBLISHED INFORMATION[1] concerning draft and temperature conditions and friction losses has been limited for the most part to large stacks or chimneys such as those used in industrial power plants. In these larger systems reliable information about stack performance is obviously desirable because of the considerable financial savings that may be involved, and large scale investigations are more practical for industrial organizations than for individuals who operate small plants or heating systems. Even for larger chimneys the experimental data available are limited.

Several types of chimneys of domestic sizes have been constructed and their performance observed at the National Bureau of Standards for the reason that experimental data on small chimneys were not available. The work was done in connection with the low cost housing program since the subject is important in that field.

It is the object of this paper to present the more important data gathered on brick chimneys. Time has not permitted any exhaustive theoretical analysis of the results of the tests but the results are presented for their inherent interest.

When tests for brick chimneys were being considered, it was planned to construct one or more such chimneys and to supply them with air heated electrically in lieu of actual products of combustion from a fire in order to minimize errors of observation and computation. There was a difficulty, however, in obtaining electrical equipment with which to accomplish this and some consideration indicated that the method finally selected would be satisfactory.

This method consisted essentially in heating air for supply to the chimney with a gas flame in an insulated fire box so that substantially all the heat of combustion was carried to the chimney by the air. The air passing through the system was measured by means of a large gas meter and the gas consumed was measured with a small gas meter so that the rate of mass flow was obtained directly.

Since the resistance to flow in the large meter was considerable, a blower and motor were necessary to overcome it and to cause a sufficient air flow through the meter.

The intent was to determine experimentally the performance of the chimney without reference to any particular kind of heating device and for this reason the flue gas temperature and available draft were measured in the flue pipe close to the chimney.

It was desired to cover the range of operating conditions likely to occur in domestic chimneys, so tests were planned during which chimneys would be supplied gases at 200, 400, 600, 800 and 1000 F. The rates of flue gas production resulting from the combustion of ½, 1 and 1½ gal of fuel oil per hour with 8 per cent and with 10 per cent CO_2 were calculated so that the available draft of a typical chimney could be observed when it was receiving gases at these rates and at the stated temperatures. The work was based on oil consumption for convenience because oil burners are commonly rated in gallons per hour. A correction would obviously be necessary if the data were used for coal burning devices since the gas evolution for the same heat release is somewhat different for that fuel.

The tests were purposely made at times when there was little or no wind in order to avoid that complication.

Data tabulated herewith show the rate at which gases were supplied the chimneys during typical tests:

FUEL RATE	GPH	½		1		1½	
CO_2 in Flue Gases	Per Cent Vol.	10	8	10	8	10	8
Gas Flow	Cfm (Std.)[a]	18.4	22.5	36.8	45	55.2	67.5

[a]Corrected to 70F and 29.92 in. Hg.

For computation of the static draft the formula used was

$$D = \frac{d\,H(T_1 - T_2)}{5.2\,T_1}$$

in which D = static draft, in. water gage (W. G.)
d = mean density of atmospheric air at time of test, pounds per cubic foot.
H = effective height of the chimney, feet.
T_1 = mean temperature of gases in the chimney, degree F absolute.
T_2 = temperature of air surrounding the chimney, degree F absolute

It was assumed for the purpose of this computation that the flue gas would have the same density as atmospheric air at the same temperature and pressure.

The static drafts were computed for the various conditions employed during these tests and are shown on the tables of results for comparison

*Chief, Heat Transfer Section, National Bureau of Standards.
**National Bureau of Standards.
†National Bureau of Standards.
[1]Determination of Chimney Sizes, by Alfred Cotton. (*Mechanical Engineering*, September 1923.)
Proportioning Chimneys on a Gas Basis, by Menzin. (*ASME Transactions*, 1915.)
Presented at the 48th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, January, 1942.

with the observed available draft. The results show that the difference between the observed available draft and the computed static draft was not great for the conditions of the tests. An important conclusion is therefore that the draft in a conventional domestic chimney could be computed close enough for practical purposes by means of the previous formula, neglecting friction, if the mean temperature of the gases in the chimney could be known. The results show further, however, that the temperature of the gases decreases greatly when they enter the chimney, even though the chimney had been in operation several hours. A mathematical determination of the temperature conditions in a chimney is undoubtedly complex. It is complicated by the fact that conditions in a domestic brick chimney probably never arrive at a steady state of operation. These things indicate that it is safer and simpler to arrive at the available draft under a given condition by direct experiment rather than to attempt a mathematical solution of the temperature condition in the chimney.

For these reasons the observed relations between chimney dimensions, entering gas temperature and quantity, and available draft are considered to be the more important essentials presented in this paper.

In order to compare the drafts available under the various operating conditions it was desirable to correct all observations to some standard condition and the condition chosen for this standard was an outdoor temperature of 32 F and a pressure of one standard atmosphere. The formula used for making such corrections was

$$S_2 - S_1 = O_2 - O_1$$

where S_1 is the computed static draft under the conditions of the experiment, S_2 is the computed static draft under the standard conditions, O_1 is the observed draft under the experimental conditions and O_2 is the corrected draft or draft that should have been observed if the test conditions had been standard.

This relation develops from the assumption that the friction (f) in the chimney and the gas flow are assumed to be the same under the experimental and under the standard conditions, because

$$S_1 - O_1 = f \text{ and}$$
$$S_2 - O_2 = f \text{ so that}$$
$$S_1 - O_1 = S_2 - O_2 \text{ or}$$
$$S_2 - S_1 = O_2 - O_1$$

Specimens and Test Equipment

The details of the two brick chimneys used for these tests are shown in Fig. 1. These chimneys were built of common brick, laid in one 4-in. thickness around a tile flue liner. The liner in one chimney was 9 x 9-in. nominal size and that in the other was 9 x 13-in. The measured inside dimensions of the liners were 7¼ x 7¼-in. for the smaller and 6½ x 11-in. for the larger chimney. Each chimney was 38 ft high above the concrete base and each was pierced with vents, 9-in. high and 10-in. wide, opposite each other in two sides of the chimney 15 ft above the burner level and at 5 ft intervals from there to the top. By opening the vents at any level and closing the chimney above them the effective height of the chimney could be established at selected heights less than the whole height of the chimney.

The chimneys were built adjacent to a structure with corrugated metal sides rising to a height of 10 ft but were not built into the wall of the structure. Thus, in effect the chimneys were exposed to the outside air on all four sides for their entire height. A cross section of the chimneys showing the location of the vents and smoke pipe thimble is presented in Fig. 1.

The mortar used was a Portland cement, lime and sand mixture proportioned for one part of cement and lime to about three parts of sand. Lime and cement were used in the proportions of about 2:1. The mortar was worked in well, so that there were no appreciable leaks. There was an air space between the brick and the flue liner that averaged about ½ in. across. The tile was laid one above the other without a mortar seal.

To close the chimney above the test height a sheet of metal reaching through at the top of the vents was wedged tightly across the chimney. To reduce the turbulence that would result if the flue gases were allowed to impinge against this flat surface, a sheet metal splitter was placed at the outlet as shown by the diagram in Fig. 1. Fig. 3 is a photograph of the two experimental brick chimneys showing the vents at several levels and the scaffolding from which the necessary operations on the chimneys during the investigation could be made.

Test Equipment

The heater used during the tests consisted of a radial gas burner somewhat less than 9-in. in diameter located in a 9-in. pipe. See Fig. 2. The burner was located about 18-in. above the floor. The pipe tapered above the burner to a diameter of six inches where it entered an elbow

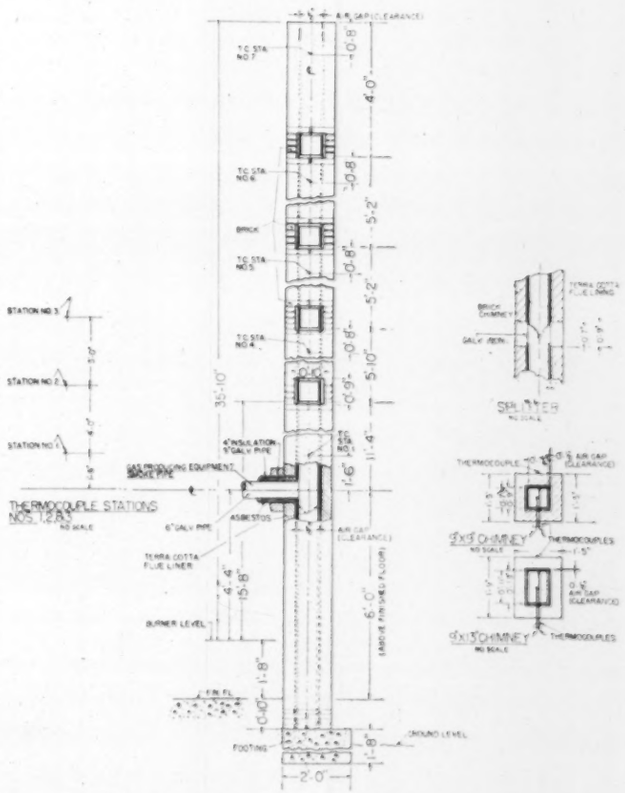


Fig. 1—Masonry chimney with terra cotta flue lining

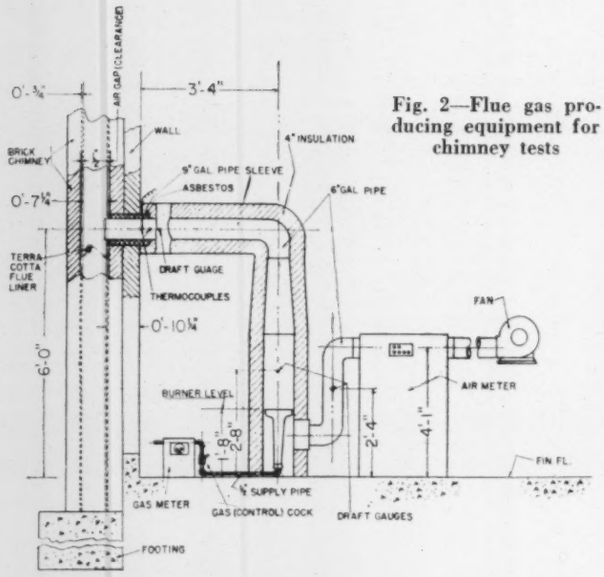


Fig. 2—Flue gas producing equipment for chimney tests

Fig. 4—Photograph of flue gas producing equipment with air meter shown in foreground

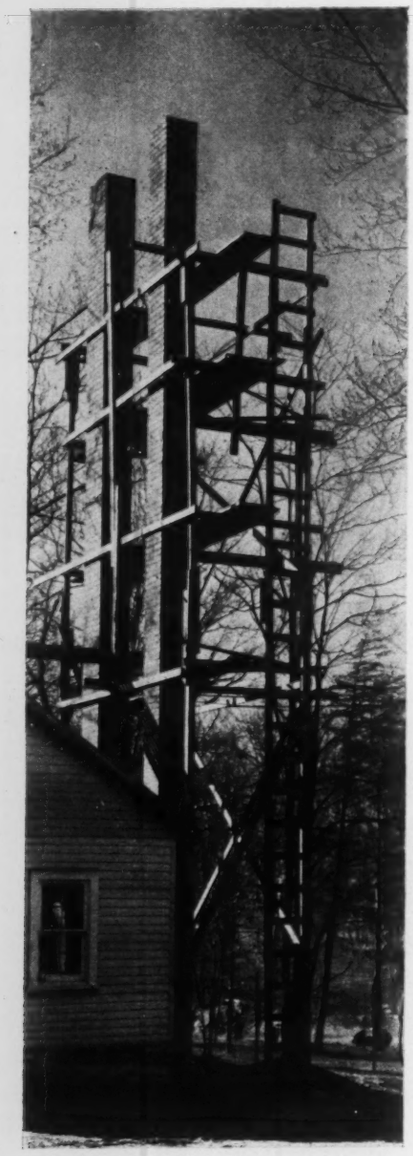


Fig. 3—View of experimental chimneys

and the elbow was connected to the chimney by a horizontal 6-in. pipe, about 4 ft long. This pipe and the thimble of the chimney were about 6 ft above the floor. The heater and smoke pipe were insulated with 4-in. of rock wool, covered with metal foil.

The draft was measured by means of a copper tube which entered the smoke pipe about 18 in. from the nearest inside surface of the chimney. The axis of the tube was normal to the flow of gases. The tube was connected to a draft gage of the inclined type, in which methyl alcohol was used as the indicating fluid and which was graduated in thousandths of an inch of water pressure. The gages were calibrated and checked occasionally against a Wahlen gage.

Temperature in the chimney and smoke pipe was observed with unshielded thermocouples made of 28 gage chromel and alumel wire. The potentiometer used was a semi-precision instrument which could easily be read to 1 F.

One pair of thermocouples was located in the smoke pipe about 18 in. from the nearest inside wall of the chimney. One of these was placed against the inner wall of the smoke pipe, the other in the center of the flue gas stream. Two thermocouples were inserted through holes in the side of the chimney at each of seven points beginning about 18 in. above the flue gas inlet and continuing at intervals of about 5 ft to the top of the chimney. In order to hold these thermocouples rigidly in position, the wires except the junctions were encased in $\frac{3}{16}$ in. porcelain insulators. The insulators of each pair were fastened together in such a way that when one of the thermocouples was pushed through the hole in the chimney until it touched the opposite wall, the other stopped at the center of the cross section. Another thermocouple at each of these levels of the chimney was held against the outer brick surface.

In some of the tests the thermocouples were moved in steps away from the chimney wall and observations were made to find the change of temperature with distance between the flue wall and the center of the gas stream. In other tests draft tubes were used to measure draft at the various levels in the chimney.

A cross section of the flue gas producing equipment and the location of the draft tubes and thermocouples stations are shown on Fig. 2. A photograph of the flue gas producing equipment is shown in Fig. 4.

It is obvious that the thermocouples with junctions placed in the axis of the chimney do not measure the average temperature of the gases, and that they are affected by radiation to the flue walls. How-

ever for the purpose of this investigation it appeared sufficiently accurate to consider the temperatures indicated by the thermocouples as representative of average gas temperatures, for such purposes as computing static draft, etc.

In the following discussion the chimneys are referred to as *15 ft chimneys, 20 ft chimneys,* etc. These heights refer to the approximate height of the chimney above the burner level. The actual heights of the chimneys were somewhat more than these nominal heights. The computed drafts given in the discussion and the tables are however based on the effective height of the chimney above the level of the draft measuring tube or center of the thimble. The relations between the effective and nominal chimney heights were:

Nominal Height, Ft	15	20	25	30	35
Height above burner level, ft	15.6	21.5	26.6	31.8	35.8
Height above draft tube, ft	11.3	17.2	22.3	27.5	31.5

General Observations

The results of the tests of the brick chimneys are shown in Tables 1 to 7 inclusive in this report.

It was observed that the temperature on the inner surface of the flue lining was in most cases somewhat lower than that in the center of the flue gas stream at the same level. There was a rapid change in temperature with horizontal distance in the layer of gases next to the wall of the flue, but except for this layer, an inch or less in thickness, the temperature across the chimney was found to approach uniformity.

In one test the temperature of the wall of the flue was 255 F, at the center of the gas stream the temperature was 367 F, while at other points one inch or more away from the flue wall temperatures varying from 351 F to 383 F were observed. In another test the flue wall and center temperatures were, respectively, 215 F and 355 F, while temperatures at other points varied from 349 F to 367 F. Other tests showed a comparatively small range of temperatures across the chimney, except in the layers near the flue wall. In general the temperature at the center was somewhat above the average, but considering the greater velocity of the gases at the center it was concluded that for practical purposes the temperature at the center of the gas stream could be accepted as representative of that of the flue gases at the same level. Accordingly, the temperatures measured at the center of the chimney were used in the computations of static draft.

The Fifteen Foot Chimney

When measured at the inlet to the chimney, the available draft in the 15 ft chimney changed from about 0.054 in. W. G. with the entering gases at 400 F, to about 0.094 in. W. G. with the entering gases at 1000 F. Very little difference in the available draft was observed when the volume of flue gases was increased in the range of gas flow rates used. When the inlet gas temperature was constant an increase in the temperature of the flue gases in the chimney was noted as the velocity of the gases was increased. This action is illustrated by data taken from the tables and presented here for convenience:

Entering Temp.	Deg Fahr	400	400	1000	1000
Flue Gas Flow	Cfm (Std.)	18.4	67.5	18.4	67.5
Gas Temp., bottom of chimney (Sta. 1)	Deg Fahr	272	300	598	680
Gas Temp., top of chimney (Sta. 3)	Deg Fahr	240	279	532	611

The curves in Fig. 5 and Table 1 show the available draft in the 15 ft chimney over a range of inlet temperatures from 400 to 1000 F for the combustion of fuel oil at various rates after continuous operation had proceeded long enough to reach steady temperature conditions. Fig. 5 also shows the equilibrium temperatures at two stations in the chimney for the various test conditions.

In each test, as noted previously, the gases entering the chimney were kept as nearly as possible at some constant temperature: 200, 400, 600, 800, or 1000 F. In the first few minutes of each test the draft increased very rapidly, usually coming nearly to a steady condition within half an hour. The temperatures at the chimney walls, however, continued to rise for about two hours. Figs. 6 and 7 show how the draft and temperature at the chimney entrance and the temperatures at the three stations in the 15-ft chimney increased with time after starting cold.

As the velocity of the flue gases in the chimney was increased, the temperature on the inner surface of the flue lining increased as well as the flue gas temperature at the center of the flue. When the flue gases entered the chimney at 400 F the surface temperature at a point 18 in. above the inlet was 205 F with 18.4 cu ft of gas flowing per minute and 258 F with 67.5 cu ft per minute. With flue gases entering at 1000 F the temperature at this point increased from 536 to 594 F, when the rate of flow was increased from 18.4 to 67.5 cu ft per minute. With a corresponding increase in velocity of flue gas the surface temperature at station 3 near the top of the chimney increased from 193 to 232 F with 400 F inlet temperature, and from 449 to 525 F with 1000 F inlet temperature.

In addition to the draft observations made at the inlet to the chimney others were made in the air duct between the large meter and the heating unit. With low rates of air delivery, 18.4 to 25 cfm, the draft at this point was slightly greater than that observed near the chimney inlet. As the volume of air supplied was increased the pressure became greater until, with a flow of 65 cfm a small positive pressure was noted. Since a positive pressure near the inlet to the heater indicates that the air is being forced into the unit, it appears that this volume exceeds the capacity of the chimney when attached to the particular heater used.

The Twenty Foot Chimney

During the tests of the 20-ft chimney changes in the weather per-

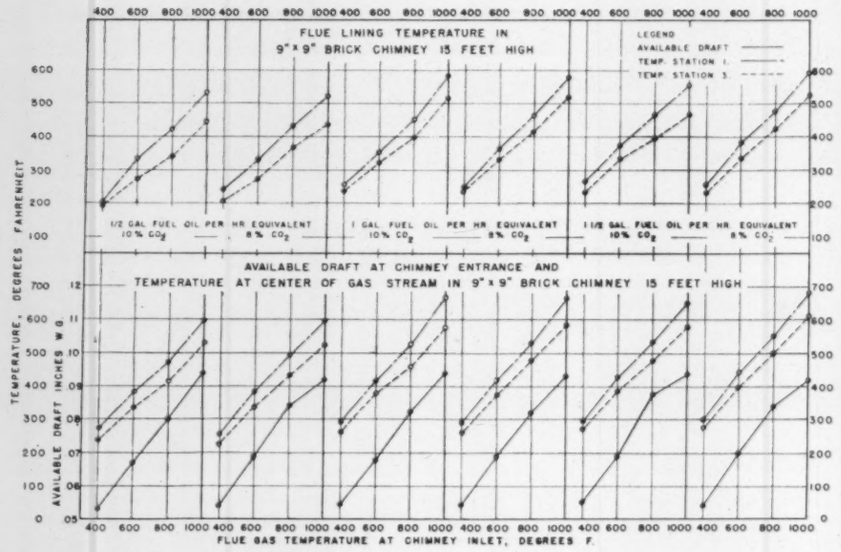

FLUE LINING TEMPERATURE IN
9"x9" BRICK CHIMNEY 15 FEET HIGH
LEGEND
AVAILABLE DRAFT
TEMP. STATION 1.
TEMP. STATION 3.
1/2 GAL FUEL OIL PER HR EQUIVALENT
10% CO2
8% CO2
1 GAL FUEL OIL PER HR EQUIVALENT
10% CO2
8% CO2
1 1/2 GAL FUEL OIL PER HR EQUIVALENT
10% CO2
8% CO2
AVAILABLE DRAFT AT CHIMNEY ENTRANCE AND
TEMPERATURE AT CENTER OF GAS STREAM IN 9"x9" BRICK CHIMNEY 15 FEET HIGH
TEMPERATURE, DEGREES FAHRENHEIT
AVAILABLE DRAFT INCHES W.G.
FLUE GAS TEMPERATURE AT CHIMNEY INLET, DEGREES F.

Fig. 5—Chimney temperatures and draft for various flue gas temperatures

mitted observation of the effect of differences in outside temperature on chimney draft. Whereas nearly all tests of the 15 ft chimney were made with the outside temperature in the range from 30 to 40 F, tests of the 20 ft chimney were obtained at 28 F, at 81 F and at intermediate temperatures. A high outside temperature lowers the density of the air and since the available draft is proportional to the difference in density between chimney gases and a lower draft results. For the sake of uniformity the values for the observed draft were corrected for an outside temperature of 32 F before plotting. A smooth curve was then obtained when the draft was plotted against the average chimney gas temperature. Corrections were also made for variation in barometric pressure. When these corrections had been made, the available draft at the standard condition was shown to change from about 0.082 in. W. G. for an inlet temperature of 400 F to about 0.135 for an inlet temperature of 1000 F.

It was observed with the 20 ft chimney height, as with some others, that the mean temperature in the chimney increased when the gas flow was increased while the temperature of the gases entering the chimney was maintained constant. The increase in mean chimney gas temperature was sufficient to cause an increase in available draft despite the greater friction loss to be expected from the greater mass flow. This effect is illustrated by a tabulation of values taken from the data:

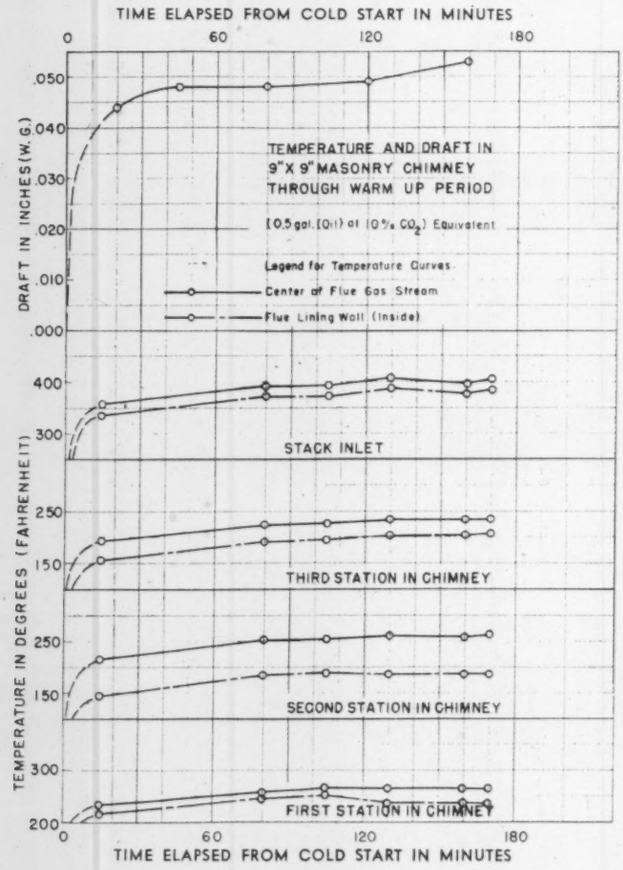

TIME ELAPSED FROM COLD START IN MINUTES
TEMPERATURE AND DRAFT IN
9"X 9" MASONRY CHIMNEY
THROUGH WARM UP PERIOD
(0.5 gal. (Oil) at (0% CO2) Equivalent
Legend for Temperature Curves
Center of Flue Gas Stream
Flue Lining Wall (Inside)
DRAFT IN INCHES (W.G.)
STACK INLET
THIRD STATION IN CHIMNEY
SECOND STATION IN CHIMNEY
FIRST STATION IN CHIMNEY
TEMPERATURE IN DEGREES (FAHRENHEIT)

Fig. 6—Temperature and draft in 15-ft 9 in. x 9 in. masonry chimney through warm up period

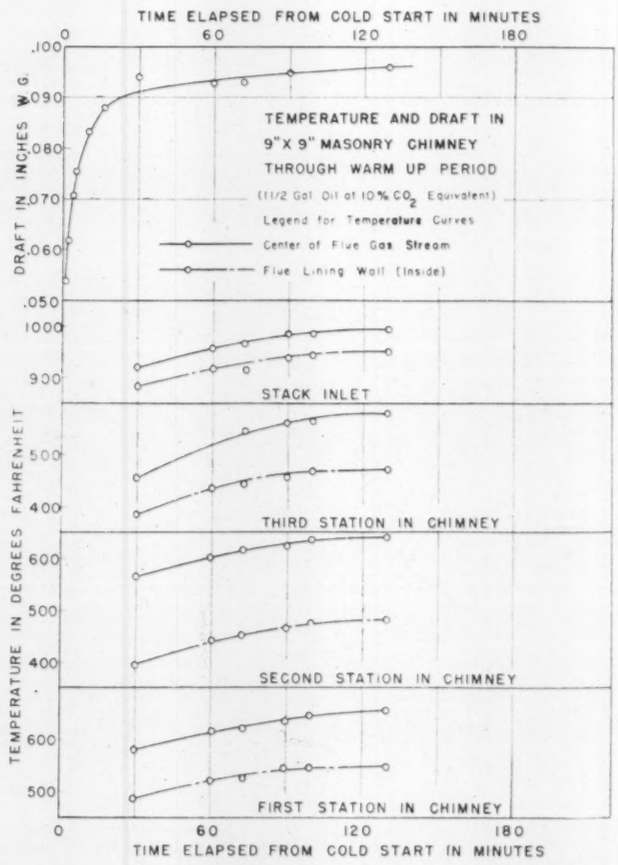

TIME ELAPSED FROM COLD START IN MINUTES
TEMPERATURE AND DRAFT IN
9"X 9" MASONRY CHIMNEY
THROUGH WARM UP PERIOD
(1 1/2 Gal. Oil at 10% CO2 Equivalent)
Legend for Temperature Curves
Center of Flue Gas Stream
Flue Lining Wall (Inside)
DRAFT IN INCHES W.G.
STACK INLET
THIRD STATION IN CHIMNEY
SECOND STATION IN CHIMNEY
FIRST STATION IN CHIMNEY
TEMPERATURE IN DEGREES FAHRENHEIT

Fig. 7—Temperature and draft in 9 in. x 9 in. masonry chimney through warm up period

Temp. Gas Entering Chimney	Deg Fahr	400		600		800		1000	
Flue Gas Flow.....	Cfm (Std.)	18.4	67.5	18.4	67.5	18.4	67.5	18.4	55.2
Temp. in Chimney, Avg..	Deg Fahr	270	297	330	425	422	555	526	645
Obs. Draft.....	In. W. G.	0.058	0.075	0.086	0.104	0.106	0.119	0.120	0.132

The average temperatures of the chimney gases shown in Table 1 are the averages of the temperatures observed at the center of the flue gas stream at Stations 1, 2, 3 and 4 shown in Fig. 1. This average temperature was used in obtaining the computed values of static draft shown in the tables.

The difference between the computed static draft, described previously, and the observed draft is an indication of the friction and inertia effects in the chimney. The results of these computations indicate that the friction loss in the chimney was not great. The difference between the computed static draft and the observed draft was less than 0.01 in. W. G. No consistent increase in friction loss was indicated in the 20 ft chimney with increased air velocities in the range of the tests.

The draft loss in the heating equipment was considerable and increased as the flow of flue gases became greater. With 18.4 cfm of flue gases passing, the draft measured 1 ft above the burner was slightly greater than that measured at the inlet to the chimney. When the rate of flow was increased to 67.5 cfm the draft at this point was about 40 per cent less than that at the inlet to the chimney.

Tables 2 and 3 contain a summary of the temperatures observed at a steady state of operation at the various stations at the center of the flue gas stream, and on the inside surface of the flue liner.

25, 30 and 35 Ft Chimneys

In view of the fact that the results of the tests of the 15 and 20 ft chimneys showed the same general characteristics, it was not believed necessary to conduct the complete series of tests for the higher chimneys. Accordingly, only the highest and lowest velocities of flue gases were used, and in the case of the 9 by 9 in. chimney the tests with 600 and 800 F entering gas temperatures were omitted. With the 9 by 13 in. chimney tests with 400 F and 800 F entering gas temperatures were omitted. The higher chimneys showed the same general characteristics as the 15 and 20 ft chimneys. Some representative data on the 9 by 9 in. chimney are presented herewith for illustration.

(1) With a constant inlet temperature, a higher average chimney gas temperature resulted when the velocity of the flue gases was increased. With gases entering the 25 ft chimney at 1000 F the average flue gas temperature increased from 504 F to 642 F, when the flow of flue gases was increased from 18.4 cfm to 67.5 cfm. In the 30 ft chimney with the same change in conditions the average temperature of the flue gases increased from 460 F to 608 F. In the 35 ft chimney, the average gas temperature increased from 488 F to 601 F.

(2) With a constant inlet temperature the draft increased when the velocity was increased, due to the higher mean chimney gas temperature. In changing from 18.4 cfm to

Table 1—Temperature and Draft in 9 in. x 9 in. Masonry Chimney[a]

Chimney Height, Ft		Fuel Oil Equivalent of Chimney Gases		Flue Gas Temp. at Inlet, F	Average Chimney Gas Temp. F	Outside Temp. F	Observed Draft Inch, W. G.	Computed Static Draft Inch, W. G.	Corrected Draft[b]		Indicated Friction Loss, Inch, W. G.
Nominal	Height Above Thimble	Fuel Oil Gal/Hr	CO_2 Per Cent						Observed Inch, W. G.	Computed Static Inch, W.G.	
35	31.5	0.5	10	236	142	68	0.040	0.053	0.076	0.089	+0.013
35	31.5	1.5	8	200	158	81	0.043	0.055	0.086	0.098	+0.012
35	31.5	0.5	10	400	224	77	0.081	0.095	0.128	0.142	+0.014
35	31.5	1.5	8	400	278	80	0.092	0.117	0.139	0.164	+0.025
35	31.5	0.5	10	1000	488	77	0.163	0.191	0.206	0.234	+0.028
35	31.5	1.5	8	1000	601	78	0.169	0.214	0.216	0.261	+0.045
30	27.5	0.5	10	250	160	83	0.043	0.049	0.070	0.076	+0.006
30	27.5	1.5	8	250	195	84	0.055	0.064	0.097	0.106	+0.009
30	27.5	0.5	10	400	213	68	0.075	0.084	0.107	0.116	+0.009
30	27.5	1.5	8	400	269	72	0.094	0.108	0.125	0.139	+0.014
30	27.5	0.5	10	1000	460	64	0.146	0.169	0.174	0.197	+0.023
30	27.5	1.5	8	1000	608	72	0.168	0.193	0.204	0.228	+0.025
25	22.3	0.5	10	250	171	92	0.034	0.037	0.072	0.076	+0.003
25	22.3	1.5	8	200	171	94	0.034	0.036	0.074	0.076	+0.002
25	22.3	0.5	10	400	243	78	0.066	0.073	0.097	0.105	+0.006
25	22.3	1.5	8	400	301	79	0.085	0.091	0.116	0.123	+0.006
25	22.3	0.5	10	1000	504	79	0.124	0.137	0.154	0.167	+0.013
25	22.3	1.5	8	1000	642	80	0.146	0.159	0.177	0.190	+0.013
20	17.2	0.5	10	250	179	94	0.029	0.032	0.059	0.062	+0.003
20	17.2	1.5	8	200	188	94	0.031	0.032	0.059	0.060	+0.001
20	17.2	0.5	10	400	270	81	0.058	0.062	0.082	0.086	+0.004
20	17.2	1.5	8	400	297	55	0.075	0.081	0.087	0.093	+0.006
20	17.2	0.5	10	600	330	50	0.086	0.091	0.095	0.100	+0.005
20	17.2	1.5	8	600	425	55	0.104	0.106	0.116	0.118	+0.002
20	17.2	0.5	10	1000	526	37	0.120	0.130	0.123	0.134	+0.010
20	17.2	1.5	8	915	588	44	0.129	0.134	0.135	0.141	+0.005
15	11.3	0.5	10	250	182	87	0.023	0.023	0.041	0.041	+0.000
15	11.3	1.5	8	200	180	84	0.025	0.024	0.042	0.041	−0.001
15	11.3	0.5	10	400	258	32	0.053	0.055	0.054	0.056	+0.002
15	11.3	1.5	8	400	293	43	0.056	0.057	0.060	0.061	+0.001
15	11.3	0.5	10	600	369	34	0.068	0.071	0.069	0.071	+0.003
15	11.3	1.5	8	600	428	40	0.072	0.076	0.075	0.078	+0.004
15	11.3	0.5	10	1000	577	32	0.094	0.092	0.094	0.092	−0.002
15	11.3	1.5	8	1000	654	55	0.092	0.090	0.100	0.098	−0.002

[a] Actual inside dimensions of flue lining 7¼ x 7¼ in.
[b] Corrected for Outside Temp. 32 F. Barometer 29.92 In. Hg.

Table 2—Temperature at Center of Flue Gas Stream in 9 in. x 9 in. Masonry Chimney[a]

Chimney Height, Ft		Fuel Oil Equivalent of Chimney Gases		Flue Gas Temp. at Inlet, F	Thermocouple Station No.						
Nominal	Height Above Thimble	Fuel Oil Gal/Hr.	CO_2 Per Cent		1	2	3	4	5	6	7
35	31.5	0.5	10	236	178	160	158	139	130	115	110
35	31.5	1.5	8	200	177	174	—	157	—	143	—
35	31.5	0.5	10	400	304	266	251	223	205	184	180
35	31.5	1.5	8	400	330	317	302	275	257	238	228
35	31.5	0.5	10	1000	669	590	550	510	488	466	425
35	31.5	1.5	8	1000	767	723	681	647	554	501	474
30	27.5	0.5	10	200	189	175	169	152	145	130	
30	27.5	1.5	8	250	215	211	206	190	182	172	
30	27.5	0.5	10	400	289	243	229	200	174	141	
30	27.5	1.5	8	400	314	312	298	272	243	237	
30	27.5	0.5	10	1000	637	542	510	427	358	276	
30	27.5	1.5	8	1000	722	701	664	585	497	479	
25	22.3	0.5	10	250	199	180	173	155	148		
25	22.3	1.5	8	200	180	177	174	152	158		
25	22.3	0.5	10	400	300	275	240	208	190		
25	22.3	1.5	8	400	341	321	306	278	260		
25	22.3	0.5	10	1000	641	586	514	426	362		
25	22.3	1.5	8	1000	735	701	664	580	528		
20	17.2	0.5	10	250	199	181	177	158			
20	17.2	1.5	8	200	179	177	175	164			
20	17.2	0.5	10	400	290	249	241	219			
20	17.2	1.5	8	400	319	299	292	274			
20	17.2	0.5	10	600	393	334	314	282			
20	17.2	1.5	8	600	438	427	416	387			
20	17.2	0.5	10	1000	632	526	480	425			
20	17.2	1.5	8	915	675	617	598	536			
15	11.3	0.5	10	250	200	177	170				
15	11.3	1.5	8	200	182	180	177				
15	11.3	0.5	10	400	272	263	240				
15	11.3	1.5	8	400	300	300	279				
15	11.3	0.5	10	600	382	387	338				
15	11.3	1.5	8	600	443	440	402				
15	11.3	0.5	10	1000	598	602	532				
15	11.3	1.5	8	1000	680	672	611				

[a] Actual inside dimensions of flue lining 7¼ x 7¼ in.

Table 3—Temperature of Inner Flue Lining Surface in 9 in. x 9 in. Masonry Chimney[a]

Chimney Height, Ft		Fuel Oil Equivalent of Chimney Gases		Flue Gas Temp. at Inlet, F	Thermocouple Station No.					
Nominal	Height Above Thimble	Fuel Oil Gal/Hr.	CO_2 Per Cent		1	2	3	4	5	6
35	31.5	0.5	10	236	175	144	138	127	108	102
35	31.5	1.5	8	200	172	156	—	136	—	132
35	31.5	0.5	10	400	291	248	219	193	160	150
35	31.5	1.5	8	400	326	292	268	239	203	203
35	31.5	0.5	10	1000	658	580	510	474	390	286
35	31.5	1.5	8	1000	761	663	602	546	435	399
30	27.5	0.5	10	200	156	139	144	139	135	129
30	27.5	1.5	8	250	189	192	174	172	162	167
30	27.5	0.5	10	400	221	215	184	173	147	
30	27.5	1.5	8	400	276	272	236	237	206	204
30	27.5	0.5	10	1000	519	475	380	357	292	162
30	27.5	1.5	8	1000	625	611	519	493	418	402
25	22.3	0.5	10	250	159	165	147	142	127	
25	22.3	1.5	8	200	164	165	—	152	140	
25	22.3	0.5	10	400	239	221	183	188	157	
25	22.3	1.5	8	400	294	280	242	249	214	
25	22.3	0.5	10	1000	536	475	369	367	300	
25	22.3	1.5	8	1000	637	611	507	505	418	
20	17.2	0.5	10	250	163	168	143	144		
20	17.2	1.5	8	200	164	165	148	153		
20	17.2	0.5	10	400	238	243	215	188		
20	17.2	1.5	8	400	306	278	249	269		
20	17.2	0.5	10	600	339	321	272	236		
20	17.2	1.5	8	600	362	399	351	320		
20	17.2	0.5	10	1000	545	497	417	330		
20	17.2	1.5	8	915	559	579	503	439		
15	11.3	0.5	10	250	161	166	139			
15	11.3	1.5	8	200	167	168	151			
15	11.3	0.5	10	400	255	196	193			
15	11.3	1.5	8	400	258	242	232			
15	11.3	0.5	10	600	333	268	273			
15	11.3	1.5	8	600	383	361	339			
15	11.3	0.5	10	1000	536	449	449			
15	11.3	1.5	8	1000	594	581	525			

[a] Actual inside dimensions of flue lining 7¼ x 7¼ in.

Table 4—Temperature and Draft in 9 in. x 13 in. Masonry Chimney[a]

Chimney Height, Ft		Fuel Oil Equivalent of Chimney Gases		Flue Gas Temp. at Inlet, F	Average Chimney Gas Temp. F	Outside Temp. F	Observed Draft Inch W. G.	Computed Static Draft Inch W. G.	Corrected Draft[b]		Indicated Friction Loss, Inch W. G.
Nominal	Height Above Thimble	Fuel Oil Gal/Hr.	CO_2 Per Cent						Observed Inch W. G.	Computed Static Inch W. G.	
35	31.5	0.5	10	204	125	78	0.035	0.034	0.078	0.077	−0.001
35	31.5	1.5	8	206	155	82	0.047	0.052	0.093	0.098	+0.005
35	31.5	1.5	8	600	353	84	0.140	0.141	0.193	0.194	+0.001
35	31.5	0.5	10	600	272	78	0.104	0.117	0.152	0.165	+0.013
35	31.5	0.5	10	1000	438	77	0.163	0.178	0.205	0.221	+0.015
35	31.5	1.5	8	1000	542	84	0.189	0.201	0.236	0.247	+0.012
30	27.5	0.5	10	260	155	83	0.040	0.045	0.081	0.086	+0.005
30	27.5	1.5	8	202	150	83	0.048	0.047	0.089	0.088	−0.001
30	27.5	0.5	10	605	278	80	0.089	0.103	0.129	0.144	+0.014
30	27.5	1.5	8	595	278	72	0.130	0.144	0.161	0.175	+0.014
30	27.5	0.5	10	1007	438	91	0.135	0.148	0.179	0.191	+0.013
30	27.5	1.5	8	995	609	89	0.166	0.165	0.225	0.227	+0.002
25	22.3	0.5	10	258	155	86	0.034	0.034	0.069	0.069	0.000
25	22.3	1.5	8	200	160	86	0.039	0.037	0.074	0.072	−0.002
25	22.3	0.5	10	600	312	86	0.086	0.090	0.122	0.126	+0.004
25	22.3	1.5	8	592	382	83	0.106	0.110	0.140	0.144	+0.004
25	22.3	0.5	10	1007	462	85	0.120	0.127	0.154	0.161	+0.007
25	22.3	1.5	8	1007	607	84	0.151	0.153	0.184	0.185	+0.002
20	17.2	0.5	10	280	175	79	0.030	0.028	0.062	0.060	−0.002
20	17.2	1.5	8	250	192	81	0.044	0.041	0.069	0.066	−0.003
20	17.2	0.5	10	598	324	74	0.069	0.079	0.089	0.099	+0.010
20	17.2	1.5	8	604	410	74	0.090	0.094	0.111	0.115	+0.004
20	17.2	0.5	10	1015	496	79	0.094	0.107	0.115	0.128	+0.013
20	17.2	1.5	8	1000	645	74	0.122	0.127	0.140	0.147	+0.005
15	11.3	0.5	10	270	172	88	0.022	0.021	0.041	0.040	−0.001
15	11.3	1.5	8	200	166	85	0.022	0.020	0.040	0.038	−0.002
15	11.3	0.5	10	604	356	85	0.035	0.032	0.073	0.070	−0.003
15	11.3	1.5	8	599	431	84	0.066	0.062	0.082	0.078	−0.004
15	11.3	0.5	10	1000	560	82	0.074	0.066	0.082	0.091	+0.008
15	11.3	1.5	8	1000	682	85	0.083	0.082	0.101	0.100	−0.001

[a] Actual inside dimensions of flue lining 6½ x 11 in.
[b] Corrected for Outside Temp. 32 F, Barometer 29.92 In. Hg.

Table 5—Temperature at Center of Flue Gas Stream in 9 in. x 13 in. Masonry Chimney[a]

Chimney Height, Ft		Fuel Oil Equivalent of Chimney Gases		Flue Gas Temp. at Inlet, F	Thermocouple Station No.						
Nominal	Height Above Thimble	Fuel Oil Gal/Hr.	CO_2 Per Cent		1	2	3	4	5	6	7
35	31.5	0.5	10	204	163	139	122	111	105	101	102
35	31.5	1.5	8	206	181	168	159	151	144	141	142
35	31.5	1.5	8	600	471	407	374	340	313	299	283
35	31.5	0.5	10	600	415	324	289	250	225	207	197
35	31.5	0.5	10	1000	673	528	469	402	359	325	305
35	31.5	1.5	8	1000	761	635	580	513	463	432	406
30	27.5	0.5	10	260	199	172	156	143	133	127	
30	27.5	1.5	8	202	180	169	163	153	147	144	
30	27.5	0.5	10	605	413	319	281	239	217	200	
30	27.5	1.5	8	595	481	418	394	343	325	305	
30	27.5	0.5	10	1007	694	531	465	401	361	330	
30	27.5	1.5	8	995	788	652	596	523	486	450	
25	22.3	0.5	10	258	196	164	150	136	128[b]		
25	22.3	1.5	8	200	179	166	160	150	143[b]		
25	22.3	0.5	10	600	416	335	305	269	245[b]		
25	22.3	1.5	8	592	479	407	383	335	305[b]		
25	22.3	0.5	10	1007	651	495	445	379	340[b]		
25	22.3	1.5	8	1007	794	647	606	521	470[b]		
20	17.2	0.5	10	280	215	179	159	145[b]			
20	17.2	1.5	8	250	215	195	184	174[b]			
20	17.2	0.5	10	598	424	333	296	253[b]			
20	17.2	1.5	8	604	486	418	385	350[b]			
20	17.2	0.5	10	1015	665	514	436	370[b]			
20	17.2	1.5	8	1000	791	661	596	540[b]			
15	11.3	0.5	10	270	187	169	160[b]				
15	11.3	1.5	8	200	179	163	155[b]				
15	11.3	0.5	10	604	433	341	295[b]				
15	11.3	1.5	8	599	507	419	368[b]				
15	11.3	0.5	10	1005	681	528	470				
15	11.3	1.5	8	1004	833	663	580				
15	11.3	0.5	10	1000	715	524	450				
15	11.3	1.5	8	995	824	653	605				

[a] Actual inside dimensions of flue lining 6½ x 11 in.
[b] Thermocouples not in place. Temperatures were estimated from readings of previous tests.

Table 6—Comparison of Performance of 9 in. x 9 in. and 9 in. x 13 in. Masonry Chimneys[a]

Chimney Height, Ft.		Fuel Oil Equivalent of Chimney Gases		Flue Gas Temp. at Inlet, F	Observed Draft Corrected[b] Inch W. G.		Average Chimney Gas Temp. F		Temp. Drop at Chimney Entrance F		Temp. Drop in Chimney F	
Nominal	Height Above Thimble	Fuel Oil Gal/Hr.	CO_2 Per Cent		9 in. x 9 in.	9 in. x 13 in.	9 in. x 9 in.	9 in. x 13 in.	9 in. x 9 in.	9 in. x 13 in.	9 in. x 9 in.	9 in. x 13 in.
35	31.5	0.5	10	250	0.076		142		58		68	
35	31.5	0.5	10	200		0.078		125		41		61
35	31.5	1.5	8	200	0.086	0.093	158	155	23	25		39
35	31.5	0.5	10	400	0.128		224		96		124	
35	31.5	1.5	8	400	0.139		278		70		102	
35	31.5	0.5	10	600		0.152		272		185		188
35	31.5	1.5	8	600		0.193		355		139		218
35	31.5	0.5	10	1000	0.206	0.205	488	438	331	325	344	368
35	31.5	1.5	8	1000	0.216	0.236	601	542	233	239	296	355
30	27.5	0.5	10	250	0.070	0.081	160	155	11	61	59	72
30	27.5	1.5	8	200		0.089		159		22		36
30	27.5	1.5	8	250	0.097		195		37		41	
30	27.5	0.5	10	400	0.107		213		111		148	
30	27.5	1.5	8	400	0.125		269		86		77	
30	27.5	0.5	10	600		0.129		278		192		213
30	27.5	1.5	8	600		0.161		378		114		176
30	27.5	0.5	10	1000	0.174	0.179	460	438	363	313	361	364
30	27.5	1.5	8	1000	0.204	0.225	608	609	278	207	243	338
25	22.3	0.5	10	250	0.072	0.069	171	155	51	62	51	68
25	22.3	1.5	8	200	0.074	0.074	171	160	20	21	22	36
25	22.3	0.5	10	400	0.097		243		100		110	
25	22.3	1.5	8	400	0.116		301		59		81	
25	22.3	0.5	10	600		0.122		312		184		171
25	22.3	1.5	8	600		0.140		382		113		174
25	22.3	0.5	10	1000	0.154	0.154	504	462	359	356	259	311
25	22.3	1.5	8	1000	0.177	0.184	642	607	265	213	207	324
20	17.2	0.5	10	250	0.059	0.062	179	175	51	65	41	70
20	17.2	1.5	8	250		0.069		192		35		41
20	17.2	1.5	8	200	0.059		188		21		15	
20	17.2	0.5	10	400	0.082		270		110		71	
20	17.2	1.5	8	400	0.087		297		81		45	
20	17.2	0.5	10	600	0.095	0.089	330	324	207	174	111	171
20	17.2	1.5	8	600	0.116	0.111	425	410	142	118	71	136
20	17.2	0.5	10	1000	0.123	0.128	526	496	368	350	203	295
20	17.2	1.5	8	1000		0.147		645		219		251
20	17.2	1.5	8	915	0.135		588		240		119	
15	11.3	0.5	10	250	0.041	0.041	182	172	50	83	30	27
15	11.3	1.5	8	200	0.042	0.040	180	166	18	21	5	24
15	11.3	0.5	10	400	0.054		258		128		32	
15	11.3	1.5	8	400	0.060		293		100		21	
15	11.3	0.5	10	600	0.075	0.073	369	356	218	171	44	138
15	11.3	1.5	8	600	0.069	0.082	428	431	157	92	41	139
15	11.3	0.5	10	1000	0.094	0.082	577	560	402	324	66	211
15	11.3	1.5	8	1000	0.100	0.101	654	682	320	271	69	253

[a] Data assembled from Tables 1-5 inclusive.
[b] Corrected to Outside Temp. 32 F, Barometer 29.92 In. Hg.

Table 7—Loss of Draft by Friction in 9 in. x 9 in. Masonry Chimney

Fuel Oil Equivalent of Chimney Gases		Flue Gas Temp. at Inlet, F	Observed Draft Inch, W. G.		Observed Draft[a] Difference Inch, W.G. Station 1-7	Computed Static Draft Inch, W. G.		Static Draft Difference Inch, W.G. Station 1-7	Draft Loss Inch W. G.	Per Cent of Computed Static Draft Lost by Friction
Fuel Oil Gal./Hr.	CO_2 Per Cent		Station 1[a]	Station 7		Station 1	Station 7			
0.5	10	200	0.043	0.007	0.036	0.045	0.001	0.044	0.008	18
1.5	8	200	0.041	0.004	0.037	0.052	0.000	0.052	0.015	29
0.5	10	400	0.073	0.008	0.065	0.088	0.001	0.087	0.014	16
0.5	10	1000	0.151	0.012	0.139	0.176	0.003	0.173	0.034	20
1.5	8	1000	0.174	0.024	0.150	0.208	0.005	0.203	0.053	26

[a] Vertical distance between stations 1 and 7 was 29 ft — 4 in.

67.5 cfm the corrected observed draft in the 25 ft chimney, with 1000 F inlet temperature, increased from 0.154 in. W. G. to 0.177 in. W. G. In the 30 ft chimney the draft increased from 0.174 in. W. G. to 0.204 in. W. G.; in the 35 ft chimney it increased from 0.206 to 0.216 in. W. G.

(3) In the 15 and 20 ft chimneys, the change in the indicated friction loss as the gas temperature was increased was smaller than the uncertainty in the data, but with the 25 ft and higher chimneys, there was a definite indication of such increase.

(4) In the tests of the 25 ft chimney as in the lower chimneys the draft loss due to friction was not much larger for the higher flue gas velocities used than for the lower velocities used during the tests. With the 30 ft and 35 ft chimney a measurable increase in the draft loss due to friction was noted for the higher flue gas velocities.

The observed draft in the upper portions of the higher chimneys was found to be greater than the computed static draft. In one case five feet from the top, the draft observed was 0.005 in. W. G. more than the computed draft at the same level and 8 in. below the top it was 0.007 in. W. G. more. In other tests at different temperatures the draft observed 8 in. below the top of the chimney exceeded the computed draft by 0.005 to 0.02 in. W. G. This seemed to indicate a chimney effect above the actual top of the chimney. Some measurements were made which indicated the existence of draft more than 2 ft above the top of the chimney. The existence of the *extra* chimney effect of the column of hot gases over the chimney complicates calculation of the draft lost by friction. However, some general conclusions may be reached by comparison of the computed and observed drafts at different levels in the chimney. If the draft observed at one point is subtracted from that observed at some lower point the result is the draft produced between those points. The draft can be calculated and a figure for draft loss in a section of the chimney can be obtained. In this way the draft loss between Stations 1 and 7 in the chimney was calculated for five tests. The maximum friction loss indicated in this way was 29 per cent of the computed static draft. This loss was obtained with a flow of 67.5 cfm with flue gases entering the chimney at 200 F. The losses of draft for other tests determined in this way are given in Table 7.

This investigation was intended to supply experimental data sufficient to answer a definite question, namely; if a known quantity of flue gas at a known temperature flows into a particular chimney, what will the chimney draft be? Two sizes of chimneys, a number of chimney heights, several rates of gas flow and several temperatures at the inlet to the chimney, were used, covering most of the combinations of conditions for which an answer might be desired. The essential measurements were therefore those of rate of gas flow and gas temperature and draft at the chimney inlet. Numerous other measurements of draft and temperature at other points were made, but these are distinctly subsidiary though they may prove useful if a more detailed analysis of the data should be attempted.

DISCUSSION

H. F. Randolph, Utica, N. Y.: I would like to inquire if any comparative data are available for lined and unlined chimneys?

Mr. Dill: Not at the present time, but as far as friction is concerned, I believe the statements reported in the paper will hold, because the chimney velocities are not great enough to have any bearing upon the roughness of a masonry chimney. Of course, if a chimney were a quarter or half constructed, there would be a measurable effect.

Question: Was any attempt made to determine the amount of leakage into the chimney?

Mr. Dill: No leakage determinations were made, but the chimney was built by a contractor familiar with bricklaying and it was assumed that this chimney was a typical job. The lining tile were placed together with mortar, but they were not flushed.

R. K. Thulman, Washington, D. C.: May we conclude from this paper that, in order to obtain a draft sufficient to operate, for example, a space heater requiring 0.06 in. draft, a masonry chimney of 15 ft is inadequate?

Mr. Dill: The answer to this question depends a great deal on the operation of the unit and whether the chimney stays warm. If a space heater is connected to the chimney with a pot type vaporizing oil burner and it is operated for a period of less than a half an hour, the probabilities are there will be a period of smoking, because the device will operate with a draft that is too low for its proper functioning. Some of these units operate with 0.06 in. of draft and others demand 0.08 in., so under such circumstances the brick chimney has the disadvantage of great heat capacity, which requires some time for the proper chimney temperature to be attained.

Mr. Thulman: May I add to my previous question the assumption that the flue gases entering the chimney from the space heater are less than 400 F?

Mr. Dill: In this case I do not believe sufficient draft would be available and under such circumstances a blower on the burner would be desirable.

C. E. Olsen, Elyria, O.: It seems to me that these tests have been made under rather unfavorable conditions with the four sides of the chimney entirely exposed. In practice, an outside chimney generally has at least one side facing an inside temperature of 70 F.

Mr. Dill: I do not believe that this was a serious matter, for the reason that the draft inside the chimney and the temperature on the inside of the walls came to equilibrium in about half an hour, whereas, the outside temperature never approached equilibrium.

Question: It seems to me that this testing arrangement placed the chimney

and the fan in series and I wonder how you compensated for that? By increasing the speed of the fan, did you not produce a pressure at the base of the chimney?

MR. DILL: Insofar as the chimney is concerned it does not know whether there is a motor, a fan, a meter or simply a damper at the point where the gases enter. As long as the desired draft is produced at a point approximating the place where the hot gases are liberated from the heating unit, the desired performance conditions will be maintained.

MR. THULMAN: Will you please make a few remarks about the flue gas temperature in long smoke pipe connnections.

MR. DILL: I recall an installation of a space heater connected to a chimney burning about a gallon of oil an hour. The temperature of the flue gases leaving the heater was 1,050 F. When the gases entered the chimney after passing through two lengths of stove pipe and elbow, the flue gas temperature had dropped to 575 F. This brings up the question as to whether the space heater should be credited with the heating effect produced from the smoke pipe.

ALLEN JOHNSON, Primos, Pa.: Have you any comparisons between vitreous enameled stacks, metal stacks and brick stacks, insofar as the amount of draft each is capable of producing?

MR. DILL: Some data are available on this subject. The amount of draft available from a metal stack depends entirely on how well it is insulated. There is a large drop in temperature of the gases upon entering such a stack, because its heat capacity is low. This situation might be an advantage in starting a fire with an oil-burning device without a forced-draft fan.

WILLIAM RITCHIE, HEATING ENGINEER, DIES AT 73

William Ritchie, Jersey City, vice-president and treasurer of the Boynton Furnace Co., New York, for 20 years before the firm was dissolved in 1935, died February 19, 1942, at the age of 73 in the Institute of Ophthalmology of the Columbia-Presbyterian Medical Center, New York. Mr. Ritchie was born in New York, April 26, 1869, where he attended public schools, graduating from Hasbrook Institute, Jersey City, and Cooper Union, New York.

Mr. Ritchie joined the Society in 1909 and became a Life Member in 1939.

Mr. Ritchie was in the employ of the Boynton Furnace Co. for more than 40 years, serving in the capacity of mechanic, salesman, office manager, superintendent, secretary, and finally as treasurer and vice-president. He was treasurer of the Masonic Lodge of the Temple, Jersey City, for 40 years, past district deputy of the Grand Lodge of New Jersey, as well as a member of the Odd Fellows, and secretary of the local chapter of the ASHVE.

He is survived by his widow, the former Eleanor J. MacLean; two daughters, Mrs. E. R. Lindsey, Downers Grove, Ill., and Mrs. Dorothy Sykes, New York, and two grandchildren, Jean Ellen and William Ritchie Lindsey, to whom the Officers and Council of the Society extend their sincere sympathy.

CHARLES H. MORTON DIES AT GRAND RAPIDS

The Western Michigan Chapter regretfully announces the death of Charles H. Morton on February 19, 1942, a member of the Chapter for over 10 years, serving

Charles H. Morton

as Treasurer of the Chapter for 1939-40.

Mr. Morton was born at Pemberton, N. J., November 30, 1863, received his early education in New Jersey, leaving school at the age of 15 to work in the D. F. Gibbs general merchandise store as a clerk. In 1904, he became inspector and supervisor of installations at Warren Webster office at Camden, N. J. He left there in 1909 and became superintendent for Hunt Brothers, heating contractors in Greensboro, N. C., and in 1914 he was salesman for J. D. Swartwout Co., Saginaw, Mich., which position he held until 1931. He then became associated with Knight Randall, Detroit, Mich., representing Warren Webster & Co., Kewanee Boiler Corp., and Ric-wil Co.

Mr. Morton numbered a great many friends among the heating trade and was recognized as one who understood the steam heating business. He was always in good health and active in business until his illness which began last May. He will be missed by his friends in the heating industry, and especially by his fellow workers in the Chapter.

The Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS extend their sincere sympathy to his widow, Mrs. Clara B. Morton, Grand Rapids; his son, Herman, Toledo, O.; a grandson, Charles Morton, also of Toledo; a granddaughter, Mrs. Ruth Metzler, Philadelphia, Pa.

The ASHVE in the War—

By S. H. Downs

[Concluded from p. 251]

ter affairs. Every member can help by seeking out new members, who will contribute by the mere act of joining, and will contribute additionally by joining the chapter. The effect of joining this society is largely hidden in the statistics of overall accomplishment. However, the effect from a member becoming active in Society affairs is a second step to increased effort and becomes evident to all.

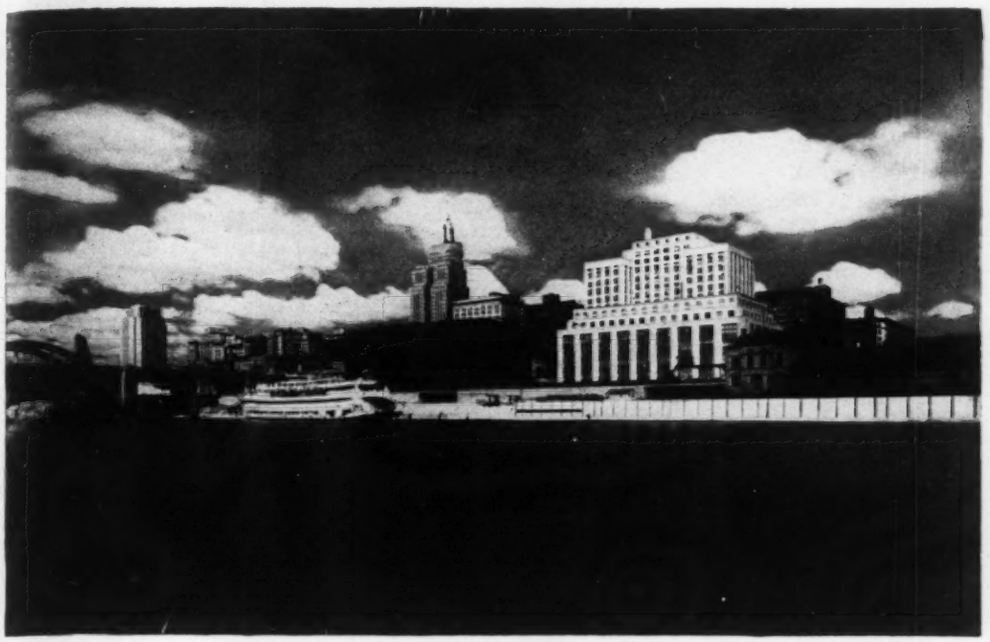
It appears, then, that the objects of the Society and their means of accomplishment are as sound during this period of war as they are in peacetime and the membership can discharge its obligation properly only if we prosecute and expand our activities to the end that we really do "advance the arts and sciences of heating, ventilating and air conditioning" by "holding meetings, issuing publications, coordinating and conducting fundamental research work, and encouraging the production of data, reports, standards, and codes."

We should encourage the expansion and the development of research and knowledge since we must look forward to a long war, and to a further period of difficult readjustment following the war, when all our knowledge will be needed to cope with our problems.

The officers and council of the Society recognize their responsibilities in this situation and have offered the research facilities of our Society to the United States in the following resolution, adopted at the 48th Annual Meeting in Philadelphia, January 29, 1942:

WHEREAS, it is the unanimous belief of the Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS that an all-out effort in the National emergency should apply throughout the country.

THEREFORE, BE IT RESOLVED, That the Society offer without expense to the people of the United States through its Government, the full use of ASHVE Research facilities including the specially trained personnel of its Laboratory.



Saint Paul's Skyline from Mississippi River

SAINT PAUL IN JUNE

June 15-17, 1942

CIRCLE THE dates of June 15, 16 and 17 on your calendar for the Saint Paul meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and make plans now to attend. This is the message of the Minnesota Chapter of the Society, which has been looking forward for many years, since the 1930 Meeting in Minneapolis, in fact, to be host for the Semi-Annual Meeting. An enthusiastic Committee on Arrangements has been working hard to perfect a program of entertainment, sightseeing and inspection which they feel will be enjoyable and beneficial to all the members and visitors who attend. The detailed program will be announced in a later issue.

Readily accessible by all modes of transportation from every section, Saint Paul, together with its sister city, Minneapolis, is at the head of navigation on the Mississippi River. As the headquarters for three major railroads and a transcontinental airline, and with six other railroads serving Minnesota's capital city, Saint Paul has frequent daily and night airliner schedules, five of the most modern streamlined trains and many other daily trains. Chicago is only six hours away, Kansas City seven hours and St. Louis just overnight. All of the day trains are timed to connect with important overnight trains to all sections of the country.

Unusual natural beauty given to Saint Paul by river bluffs, white sandstone cliffs, rolling hills and a great horseshoe curve of the Mississippi River make the city a place of scenic interest, particularly to summer visitors. Saint Paul has eighty-eight improved park areas which contribute greatly to the delightful recreational opportunities to the visitor. Outstanding among these is Como Park, known as one of the most beautiful parks in the entire country, resplendent with flowers, trees and shrubbery, skillfully arranged. The conservatory houses a large collection of interesting flowers and plants from all parts of the world, and many outstanding flower shows are held there each year.

Many beautiful drives encircle and intersect the city. Most famous of these parkways is the River Boulevard extending along the high wooded bluffs of the Mississippi River. This boulevard forms a part of the extensive parkway system encircling Saint Paul and connecting the many fine municipal parks.

Eleven college campuses and many magnificent public buildings and churches, many of which may be seen along the parkway system, give added interest to a tour of Saint Paul. The Committee on Arrangements for the Semi-Annual Meeting urges the members of the Society attending the Minnesota meeting to come to Saint Paul in June.

COMMITTEE ON ARRANGEMENTS

R. E. Backstrom, *General Chairman;* A. B. Algren, *Vice-Chairman. Executive Committee:* N. D. Adams, H. E. Gerrish and F. B. Rowley. *Banquet:* H. M. Betts, *Entertainment:* E. F. Jones. *Finance:* M. S. Wunderlich. *Golf:* E. F. Bell. *Ladies:* Mrs. R. E. Backstrom. *Publicity:* D. B. Anderson. *Reception:* M. H. Bjerken. *Transportation:* J. F. Stafford.

Como Park Conservatory

Proceedings of Annual Meeting 1942

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

(Concluded)

FOURTH SESSION—

Tuesday, January 27, 9:30 A.M.

The fourth session was called to order by President Fleisher, and W. S. Harris presented the first paper Performance of a Hot-Water Heating System in Research Home, by A. P. Kratz, M. K. Fahnestock, W. S. Harris and R. J. Martin (complete paper published in December 1941 ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*).

J. C. FITTS (WRITTEN): This paper, while it deals with a type of system with which all of us are familiar, is of fundamental importance because in it we are dealing with integrated results. In the past we have established a number of interesting, important, separate facts and used them as the foundation of our method of design. We have tested radiators and boilers to determine what they would do in a heating system. We have found the friction losses in pipes. We have determined the heat transmission through various types of wall construction. We know a great deal about the various parts of a heating system and the conditions we are liable to meet in different kinds of buildings.

Now our research is taking a fundamental departure from the study of parts and we are to have a careful study of the over-all operation of a radiator heating plant under actual service conditions, yet with facilities for accurate observation and scientific control. As a check upon our practice this should prove very valuable.

An interesting feature of the report is the demonstration of our ability to distribute heat evenly throughout a home. The small temperature differences within the rooms and between the various rooms confirms the validity of our methods of design. This would lead one to the belief that we have enough knowledge of isolated facts to do our job well and that now our research should be directed toward the solution of problems arising from the study of complete systems in complete structures which after all is the environment in which the human being has his existence. Equally important is the study of the reaction of the individual and of groups of individuals to this environment which we help to create.

These studies may uncover the need for further research in connection with parts of the system but it will then remain subordinate to the main research projects. It is to be hoped that this work will be continued.

A. G. DIXON, Racine, Wis. (WRITTEN): Under Results of Tests, comparison of series *A* and series *B* oil burner control method brings out distinct economy advantage (approximating 10 per cent in fuel plus power costs). For series *B* method; this is to be expected, since series *B* control procedure results in low boiler water temperature when outdoor temperatures are mild. All tests reported omit the use of heating system for provision of domestic hot water and series *B* control method would not be feasible if domestic hot water were being provided.

It is presumed that future tests will incorporate employment of facilities for generating supply of domestic hot water within the residence heating plant. It is well known that in many parts of the country a higher cost of fuel must apply if domestic hot water is heated separately from the heating plant. Even where this is not the case, stand-by losses are inevitable in separate domestic hot water heaters. Either or both of these factors might operate to negate the fuel economy differential now apparently favoring series *B* method.

While the Research Home cannot deal in the scores of possible variations of fuel and equipment costs, that might apply with various systems and in various parts of the country, further tests to include integral operation of domestic hot water heating could apparently be quite profitable.

Most interesting single unanswered question in the report is that radiation installed in the Research Home is calculated to be only enough to heat the house when radiation is continuously provided with water averaging 195 deg temperature in the radiator. Yet under (extrapolated) extreme condition of —10 F outdoor temperature, circulator operated only 12 hours per day instead of 24. It is noted that superficially this would seem to indicate that radiators are oversized for the house.

If we are prompted by this to assume that standard methods of heat loss calculations now endorsed by the Society result regularly in the provision of an excess of radiation, then we may well inquire, with particular reference to residence application, *is this good or bad?* We should not be too quick to say that it is good since it gives a factor of safety. Perhaps it is bad, in view of the tremendously increased use of automatically-fired heating systems in residences during recent years. The best general answer to the problem that we commonly term *cold seventy* discomfort, is a closer approach to continuous operation of the system, *by one or another means.* Like the cook who with good intentions doubles the amount of butter and sugar called for by the cake recipe and thereby gets a bad cake, the engineer who puts a high factor of safety into his radiation calculations may conceivably be hurting performance, rather than helping it. It seems to me that further study of this particular vein of exploration holds exceedingly interesting possibilities.

It is elementary but nevertheless worth noting that the *proportion* of loss of heat from (well insulated versus poorly insulated) recesses, compared with total loss of heat from a house, will vary according to the total character of insulation of the structure. In the Research Home excellent insulation of the structure as a whole is provided; hence any weak spot in the insulating armor of the recesses appears as its worst by comparison. In the light of this arithmetical ratio, it is a fortunate natural tendency to insulate radiator and convector recesses well, in residences that are being generally well insulated. Nothing in this comment, of course, is intended to oppose the logic of advisability of insulating radiator recesses.

More involved is the question that can be raised as to whether identical radiators to those covered by report, but *not recessed,* but rather set out in the room proper in the manner commonly called *exposed,* would give identical room temperature gradients to those experienced when the radiators are recessed, and would otherwise demand only commensurate supply of heating medium in order to satisfy the same comfort level setting of thermostat that was provided in operation where the radiators were recessed. It seems quite likely that the chimney action of recessing of the radiators will have affected the ratio of convected versus radiant heat emanating from these radiators and that this in turn could influence performance results.

What I am trying to say is that tests reported up-to-date do not permit the conclusion that recessing of radiators in any except the best-insulated recesses will, per sé, increase total fuel costs. At least it may be questioned that such fuel cost increase is subject to simple calculation based upon wall structure remaining beyond a radiator recess on the one hand, and complete wall structure behind an exposed radiator on the other hand. In short, it seems possible and even likely that another variable, one of performance of radiators according to location in recesses or exposed, applies. Perhaps those who direct its

program will be able to justify future Research Home fuel cost investigation, of a kind that will throw further light on this interesting detail.

H. F. HUTZEL, Detroit, Mich. (WRITTEN): The authors indicate in Figs. 6 and 7 that by extrapolating the circulator operating time curve, a total operating time of only 12 hours per day is obtained instead of 24. This is particularly interesting in view of the fact that the installed radiation was selected on the basis of being just sufficient to off-set the calculated heat loss at an outdoor temperature of —10 deg, and with the circulator operating continuously.

From data obtained during the test they were not able to offer an explanation for this short operating time. I am wondering if this condition could not in part be accounted for by the fact that the radiation installed was selected on a basis of 200 Btu per square foot per hour. With 195 deg entering water and 185 deg leaving water in a room of 70 deg, the coefficient of heat transmission figures out to be 1.65 Btu per degree temperature difference per hour. As I recall a 3-column large tube radiator 38 in. high has a coefficient of 1.65 under such conditions of operation. I would expect that a 19 in. high radiator, 4 tubes deep and of a small tube type, would have a considerably higher coefficient of heat transfer due to its lower height, and the fact that it possesses less inter-reflecting surface. I would expect such a radiator, under stipulated conditions of operation, to have a coefficient of heat transfer nearer 1.85 or approximately 12 per cent greater than that on a basis of which these radiators were selected. If this is correct, the installed radiator capacity would represent 51,500 Btu, which is approximately 18½ per cent in excess of the calculated heat loss.

In addition to this increased heat capacity of the radiators, we must also add a percentage of the heat lost by the piping, a large percentage of which is available for compensating for part of the calculated heat loss.

There may be one or two other influencing factors which may help to explain the short running time when operating under design conditions. They are—wind velocity at time tests were conducted, as well as, whether or not the sun was shining continuously during the day time.

I note that the burner used was fired at 1.18 gal per hour during the *on* period. Assuming that the oil used had a heat value of 140,000 Btu per gallon and assuming that the boiler and burner efficiency was 75 per cent, this would provide for a heat output at the nozzle of 115,000 Btu. This capacity is 256 per cent of the calculated heat load. One would expect that a burner of this capacity on the load in question, would short cycle and operate for a short period at a time. I cannot reconcile, however, why the number of cycles for the burner and the circulator should be so closely in accord. If these curves are extrapolated to 80 deg temperature difference they indicate that both the cycles of the burner and of the circulator would be approximately 29 cycles in 24 hours. Here again, I am surprised at the curvature of the circulator cycles. If the radiators were properly selected for the load in question one would expect the circulator to operate continuously at 80 deg temperature difference unless the water supply to the radiators should exceed 185 deg. In the case of the burner cycle curves we naturally would expect fewer cycles under full load conditions. The number of cycles, of course, would depend upon the relationship between the capacity of the burner and capacity of the radiation.

I realize that the immediate object of the tests discussed in this paper was to compare the operating characteristics under two different methods of control. This probably explains why the authors did not show a table showing rate of water flow through each radiator, temperature in and temperature out, and the temperature of water in the boiler at the cut-out point. I am wondering if there was not sufficient over rise in temperature to account for the shape of the operating performance curves above referred to.

C. J. STERMER, Chicago, Ill. (WRITTEN): The writer has observed one part of the data which might cause a reader to jump to the conclusion it would be bad economics to heat the water for domestic supply by indirect means from the house heating boiler. This does not seem to be borne out in practice. Item No. 1 in the summary lists an 11.7 per cent increase in seasonal operating cost due to the heat losses at the top of the chimney. This might be termed, *stand-by loss*. In figuring the cost of domestic water heating, account would have to be taken of the *stand-by loss*, which would occur if a separate water heater were employed. The *stand-by loss* from the separate water heater, coupled with the probability of a higher cost fuel, could account for a fuel cost that would offset the losses observed in the tests reported and might prove the indirect water heating method the least costly. Also having automatic firing devices operate throughout the year for the water heating may prove desirable from a service standpoint, rather than to have the equipment idle during the summer months.

Another point to consider is that if a separate heater were used during the winter time, there would be stand-by loss occurring from two devices, the separate water heater and the central heating plant, instead of from one.

It is suggested that in subsequent studies of water heating costs, the stand-by loss from various heaters and effects of idleness on equipment be taken into account in cost of operation comparisons.

F. E. GIESECKE, College Station, Tex. (WRITTEN): The authors state that if the circulator operating time curve of Fig. 7 were extrapolated to 80 F, a total operating time of only 12 hours, instead of 24 hours, is indicated and that this might indicate that the radiation is considerably oversize for the house.

If, in the same figure, the cycle curve were extrapolated to 80 F, it would indicate 30 cycles in 24 hours. Consequently, every heating cycle would consist on an average of 24 min of forced circulation and 24 min of gravity circulation. The radiators would dissipate heat during the 24-min period of gravity circulation as well as during the 24-min period of forced circulation, but at a lower rate since the average water temperature in the radiator would be lower. If both periods of heat dissipation are taken into account it may show that the radiation is not materially, if any, oversize for the building.

It would be very valuable to have a continuous record of the temperatures of the water leaving and returning to the boiler and also a continuous record of the volume of water flowing through the boiler, determined by means of an elbow meter in the main flow or return line.

From Fig. 12 B it appears that the cost of burner and circulator power is about 10 per cent of the cost of the fuel. It would be valuable to know what portion of this power cost applies to the circulator in order to make a study of the relative economy of a forced circulation system using circulators and an ordinary gravity circulation system.

Finally, the authors state that the average heating season in Urbana consists of 204 days at an average temperature of 38 F or of 5,500 degree-days.

Since the average indoor-outdoor temperature difference is to be 72 minus 38 or 34 F, would it not be well, at this time to propose a new unit and to say that the average heating season in Urbana consists of 204 days at an average indoor-outdoor temperature difference of 34 deg or of 6936 degree-days, based on an indoor temperature of 65 F which is only indirectly related to the heat requirements of the building.

FERDINAND JEHLE, Indianapolis, Ind. (WRITTEN): Items 1 and 2 of the summary are of particular interest. It was rather a surprise that the tests under series *A* did not show some over-run in room temperature. The greater cost of heating during series *A* is of particular interest. It touches on a subject upon which there seems to be very little published data. It is hoped that Professor Kratz will, at some future time, pub-

lish a chart showing the variation of stack losses against boiler water temperature.

Some mention is made of radiator supply water temperature but complete data are not included in the paper, as to how this varied against time on both the series *A* and series *B* tests. In mentioning this to one of the authors, he stated that much of these data had to be eliminated from the paper in the interest of space. Original data, such as these, are always of value because they make the work applicable to other problems which members may be confronted with. It is, therefore, suggested that the Society permit the publication of such valuable information, although it might occupy considerable space.

The statement is made in the paper that the maximum difference in the average temperature at the 30 in. level, observed in any two rooms, was 2.8 F. It is not clear just what was averaged but, presumably, the temperature over 24 hours. It would be of interest to know what the maximum variation was between two rooms at any one time.

MR. HARRIS: The principal points raised in the discussion dealt with radiator output, radiator water temperature, water temperature in the entire system and the amount of water circulated, most of which we have already obtained. At the present time we do not have the exact data on radiator output, but we are equipping a laboratory where radiators can be calibrated to determine whether the output is in accordance with manufacturers' catalog ratings.

At no time did we experience water temperatures in the system in excess of 200 F, which was under our supposedly determined operating condition, but it should be remembered that during these tests we did not experience any 80 F indoor-outdoor temperature differentials. This year we have had a few days in which the weather conditions got down to about an average of 0 outdoors, which is close to our design condition, but even under such conditions the average radiator temperature did not exceed 200 F. Hence, it does not seem that our radiators are over-sized except for one factor. It is apparent that we are introducing heat into the house not only from the radiators, but also from uninsulated basement piping and from the inside chimney. At the present time we have not completely analyzed these losses, but we feel that we will be able to report this information in some future paper.

It was not our intention to interpret that the heat loss from the house is actually only about half of what it was calculated, but it may be almost as great as that calculated. However, our results indicate that the heat introduced by the radiators as determined from the extrapolated circulatory performance curves indicates that the radiators are being used at about one-half capacity.

A question was raised regarding the maximum variation between rooms. All temperatures reported in this paper are for an average of a 24-hour test period and the maximum difference was 2.8 F which would be almost the same for any individual time in that we carried uniform house temperatures for the 24-hour period. There were no night set-back conditions involved in these tests and the maximum variation in the rooms was about 0.5 to 1 deg F throughout any period of an operating cycle, so there is little opportunity for greater variation than the 2.8 F temperature previously mentioned. The total power consumption represents about 10 per cent of the cost. The circulator and burner power consumption split almost equally, hence, the cost of operating the circulator represents about half the power consumption or about 50 per cent of the total cost of operation.

Vice-President Eastwood introduced Prof. W. M. Sawdon, who presented the paper, Factors Influencing the Heat Output of Radiators, by A. C. Davis, Professor Sawdon and David Dropkin (complete paper and discussion published in March 1942 ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*).

Dr. Giesecke then presented the paper, A Comparative Study of the Friction Heads in Screwed and Welded Elbows, by Dr. Giesecke and J. S. Hopper (complete paper published in January 1942 ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning.*)

E. L. WEBER, Seattle, Wash. (WRITTEN): The advantages of welded connections over screwed connections in reducing friction are shown to be much greater than generally anticipated. Add to this a great number of efficient welders now being trained and we can look forward to the reduction of the use of screwed fittings and thereby improve the design of our piping systems.

W. R. RHOTON: I would like to inquire of Dr. Giesecke what the maximum allowable water velocities are to preclude undesirable noise. I believe THE GUIDE states that water velocities of 11 fps constitute the upper limit for a building other than a factory. I was wondering whether you have observed in your laboratory work any undesirable velocities around 8 fps. Also I would like to know whether the old theory still applies that the friction head increases with the square of the velocities.

E. C. SMYERS, Pittsburgh, Pa.: In Fig. 7 of the paper the pipe sizes are shown from ⅜ to 8 in. Are these actual inside diameters or are they standard pipe sizes.

DR. GIESECKE: Answering Mr. Rhoton, we have had no experience with velocities high enough to produce noise although I have observed this factor in velocities as low as 5 fps. As to the rate at which friction heads increase with velocities, this depends on the pipe size and that in turn depends on the relative roughness. For the 2½ in. pipe, I called attention to the fact that it varied as the 1.81 power of the velocity. For the same absolute degree of roughness the relative roughness in a 10-in. pipe is less than it is in a 4-in. pipe and for that reason the exponent is lower for the former size. The sizes marked on the charts are for nominal sizes for standard pipe, but the small *d* in the former applies to the actual internal diameter of standard pipe.

The last paper on this session is entitled, Design and Performance of a Direct-Fired Unit Heater, by R. M. Rush and H. A. Pietsch, and was presented by Mr. Rush (complete paper published in January 1942 ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*).

E. K. CAMPBELL, Kansas City, Mo.: You referred to the fact that the output of a heater is approximately 6000 Btu per square foot. Do you recommend that this rated capacity be operated at all times, or is there a limiting output that you recommend from the standpoint of durability? I would also like to inquire what the operating outlet temperature of the unit is, or the temperature rise of the air as it passes through the unit.

W. M. MYLER, JR., Columbus, Ohio: Before asking a couple of questions I would like to state that there are both steel and cast-iron units available for the type of service referred to by the authors of this paper. In studying the tables of data that were presented, there were a few things that seemed a little unusual for gas-fired equipment, and that is the basis on which these questions are asked.

The carbon dioxide values shown in these tests are abnormally high as compared to what we normally consider for gas-fired equipment. For instance, values are given around 10.2 and over 11 per cent, with one case showing an oxygen content of less than 2 per cent. How is this high carbon dioxide controlled and is it a field or factory adjustment?

Two tests are shown operating at a 25 per cent difference in rating. Normally a change in rating of this amount would cause a corresponding change in the carbon dioxide and yet we did not find anything reported that would indicate such an alteration. Does this mean a change in construction of the unit or

simply a change in adjustment of the heater between the two test conditions? A reference was made to the *American Gas Association* approval which apparently applies only to the million Btu unit, but I would like to inquire whether it also applies to the higher rating.

Do the carbon dioxide values, which are much higher than we normally find in the *AGA* approved appliances, actually correspond to the figures of that same unit when it was tested and received *AGA* approval. If the answer to this question is yes, then there is another question that comes to my mind. How do you take care of the normal fluctuations in Btu content, because we know in some localities variations of natural gas exist which have extremes as high as from 1025 to 1175. Obviously, if the heater is adjusted to a gas heat content of 1025 with a 2.1 per cent carbon dioxide, it would seem trouble might develop if gas having a heat content of 1175 is introduced.

F. E. Spurney, Washington, D. C.: I would like to ask a question about the maintenance of these heaters. What is the longest period of time you have had such units installed and how did they stand up in actual practice. What are the maintenance costs?

L. E. Seeley, New Haven, Conn.: I am wondering whether the 10 F temperature gradient mentioned would apply to all outside temperatures, or was there some particular temperature at which this was determined.

Mr. Pietsch: With reference to Mr. Campbell's question, the output on the basis of 6,000 Btu per square foot is considered an average. In the upper half of the unit most of the heat transfer is by radiation, but in the lower half, which is covered by plastic material, some of the heat released is obviously obtained by a combination of conduction and radiation. Hence, the heat transfer is not as high in the lower section as it is in the upper half of the heater. We have estimated from test results that from 75 to 80 per cent of the heat is transferred in the upper half of the heater by radiation and a small amount by convection. In order to maintain the proper combustion chamber temperatures we have found that the air velocities over the heating surface run approximately 45 fps. The outlet temperature rise of the heater depends on the amount of air circulated but it runs approximately 70 to 80 F above room temperature.

The high carbon dioxide obtained can only be produced with controlled combustion, which is accomplished by two means. At the gas burner, combustion air can be controlled from the supply fans, can also be controlled by a damper in connection with the exhaust fans, and of course, by the control of the gas supply. The burner is a pre-mix type and consequently most of the air is mixed before it is fed to the burner.

The 25 per cent difference in rating is accomplished by increasing the volume of the fans. The heat transfer is increased as a result of the extra air velocity and we maintain the proper carbon dioxide by controlled combustion. The difference in 25 per cent of the rating is not accounted for exactly by a difference in construction. For the million and the million and a quarter Btu units, they are identical with exception of the amount of gas burned and the amount of air circulated over the heater.

Difference in Btu content of gas in various localities is taken care of by the proper selection of burners so that we obtain within a few cubic feet of gas, the required volume that is necessary for a particular installation. Service engineers are assigned to each installation to set the performance for the desired local condition. Obviously, high carbon dioxide maintained on test is not held in the field where we set the combustion rate around 9 to 10 per cent which will allow for a certain fluctuation of the Btu value of the gas. The tests of the *AGA* do not operate as high as the carbon dioxide mentioned in the paper, due to the fact that it is not necessary in their tests to carry such a high carbon dioxide value. Under these conditions the carbon dioxide ran approximately 9 to 10 per cent on the equipment in the test laboratory.

It is important to keep the combustion chamber cool and below the point where the metal will not oxidize. We have combustion chamber temperatures ranging from between 1600 to 1800 F.

Regarding the temperature gradient that can be maintained between the floor level and the ceiling the data reported in the paper were obtained when the outside temperature was 36 F. I cannot say whether it would be any different for a lower outside temperature, but I do not imagine it would be much greater.

FIFTH SESSION—

Tuesday, January 27, 2:00 p.m.

The fifth session was called to order by President Fleisher, who introduced Prof. G. L. Tuve, and he presented the paper, Entrainment and Jet-Pump Action of Air Streams, by Professor Tuve, G. B. Priester and D. K. Wright, Jr. (complete paper published in November 1941 ASHVE Journal Section, *Heating, Piping & Air Conditioning*).

W. E. Zieber, York, Pa. (Written): The manner in which the entrainment ratios and the total throw experimental figures conform with the calculated figures in Table 2 is very interesting. The writer believes that this paper would be more complete if the experimental data are compared directly with calculated results from the formula or if the factors were put in the theoretical formula that would make them line up with the experimental work, especially in the reduced slot widths below 3 in. The formula could be rearranged or factors applied to cover the effect when narrow slots are used that would be of considerable engineering value. It appears from the information available that there might be sufficient data to accomplish this result without going into any more experimental work.

Table 2 lists some assumed jet angles up to 90 deg and shows the effect upon total throw. Some experimental work to confirm this would be very interesting and useful to the engineering world.

Velocities at the outlet up to 2400 fpm have been used in this investigation. We would like to know whether the noise level is sufficient to eliminate these velocities in practice or whether certain openings could be used with these velocities without worrying about noise effect.

Has there been any information obtained in this work that would show how far apart two narrow slots could be placed without affecting entrainment capacity of each?

C. M. Ashley (Written): It seems unfortunate that some of the openings were such as to give an orifice effect so that the opening does not represent the true size of the stream. However, if the magnitude of this effect is known it should be possible to reinterpret the data on a satisfactory basis.

A second point which I cannot understand is the data on the momentum of air streams with different types of grilles. You would expect that for a given air quantity through a given face area the type of grille would have no appreciable effect upon the momentum, yet this does not appear to be the case from the data.

The third point concerns the data shown in Table 9 which seem to indicate:

(a) that the throw of a square duct is less than the throw of a rectangular duct with the same area. I do not believe that this is actually the case and wonder what comments the authors have on this particular point.

(b) Another point from the same table is that the ratio of throw between the small square opening and the large square opening does not seem to be proportional to the dimensions as should be expected and as our data all indicate it to be.

If, however, the throw for the large square outlet were increased to 36 ft at 800 fpm and if the throw for the small square outlet were decreased to between 22½ and 23 ft for 800 fpm, consistent with the results for the same outlet at 600 and 1200 feet and if instead of the physical size of the outlet the throw was made proportional to the square root of the momenta then the result would check. I think this is a point of some importance because it is the only quantitative data from which the effect of size can be inferred.

Perhaps the most significant comment which I might make is that the induction ratio shown in this paper does not correspond to that obtained in practical experience. The reason for this is that in an actual application there is a considerable back flow of air counter to the air stream. This back flow results in a resistance to the forward motion of the air stream, which materially slows down the forward motion close to its terminus and, at the same time, peels off the outer layers of the stream so that they can no longer be recognized as a coherent part of it.

As a result of this effect the induction ratio is limited according to the approximate relationship, ratio total volume to primary volume of the stream equals initial velocity divided by 200. Thus, if you had an initial velocity of 1000 fpm, the induction ratio would be 5 to 1.

The figures which I have given are based upon a back flow of approximately 50 fpm. Obviously with higher return velocity through the room, the induction will be still less, whereas with lower velocity the induction will be greater. Incidentally in a confined room it is quite easy to measure the maximum rate of induction since this can be done by measuring the return air velocity after deducting the cross section of the stream itself from the room sectional area.

It is to be hoped that this splendid work will be followed up by a study of the effect of temperature on both horizontal and vertical air streams and this work should include not only an experimental program but also a correlation with the theory.

C. H. Coogan and John A. Goff, Philadelphia, Pa. (Written Abstract—complete discussion to be published in ASHVE Transactions 1942): The authors present experimental data which should interest everyone concerned with the entrainment problem. The theory of the turbulent mixing of jets was first developed by Tollmein,[1] who applied Prandtl's[2] conception of apparent shear stress and mixing length to the problem. Tollmein examined theoretically three cases: (*a*) the plane parallel jet with mixing on one boundary only; (*b*) the plane jet; and (*c*) the rotationally symmetrical jet. Kuethe[3] extended Tollmein's theory to the case where the driven fluid has an initial velocity parallel to the driving fluid. The writers[4] extended the Tollmein theory to the case where the jet issues into still air and the driving and driven fluids for widely different densities.

A great many air outlets encountered in the heating, ventilating and air conditioning field can certainly be idealized to one of the cases previously mentioned for certainly the conditions of essentially constant pressure and uniform density are customarily those encountered in the field. The empirical constant must of course be determined for each similar case, but this can possibly be determined from the complete experimental data of the authors.

Förthmann[5] determined experimentally the velocity profiles for air discharging from a slot 1.18 in. by 25.6 in. with a maximum discharge velocity of 113.3 fps. The results of his experiments should check closely with the data from the authors' 1 in. x 24 in. slot, tested at 40 fps maximum velocity, when plotted on a comparable basis.

The plottings of the uncorrected velometer readings taken from Figs. 7 and 8 of the paper with smoothed curves sketched through the data, indicate considerable non-symmetry which should be explained. Is this non-symmetry inherent or was it caused by the supply conditions to the slot or by incorrect velometer readings? Was the flow steady or were there small pulsations and what effect did they, if existing, have on the velometer reading?

The authors refer to the *momentum theory* and have gone to considerable trouble to calculate and prove that the *total momentum* (ΣwV^2A) is essentially constant throughout the jet. The momentum theorem is one of the fundamental theorems of fluid mechanics. It is usually obtained by an application of Newton's laws.[6] If correctly applied, the theorem of momentum, being fundamental, will always hold. The theorem of momentum by itself will not furnish much information regarding the mixing action. The quantity which the authors call *total momentum* will obviously be constant provided: (1) the resultant pressure in the x direction on the bounding surfaces of the jet is zero; (2) there are no impressed forces such as gravity, and (3) the flow is stationary. Under these conditions the $\int\int wV^2dA$ will be constant over vertical planes throughout the jet (V is the x component of the velocity). It seems to the writers that the authors have, by showing that the calculated *total momentum* is practically constant, simply proved that the foregoing assumptions (1), (2) and (3), are essentially correct.

F. J. Kurth, New York, N. Y.: In air entrainment problems we have found that temperature of the streams has a decided influence on the action. Also the entrainment on the mass of air leaving the outlet will vary considerably. Forces of aspiration are working when part of the energy of the out-flowing air draws the air mass back into the outlet. An outlet that releases an entrainment without aspiration results in a completely different result with respect to aspect ratio, distance and velocity as compared to an outlet that has aspiration. I suggest that future research investigate the fundamental effect of aspiration on air diffusion.

Mr. Priester: With reference to Mr. Ashley's comments, we used a plenum and a flat plate outlet more for simplicity's sake, so that we could obtain data on the variables in an easier manner. We realize that more work should be done with a duct approach to an outlet. It was observed that the air stream contracted as it left the sharp edged orifice. We would like to have proved equation 2 in the paper, namely, that the throw is equal to some constant times the velocity times the square root of the dimensions, but we were unable to take enough readings on total throw to complete the entire analytical data. In Table 9, it will be observed that the first item for the square opening is out of line with the rest. This is probably the one figure which Mr. Ashley mentioned and it might possibly be that this figure should have been higher. We took only a very few throw readings. In answer to the comment on counter flow effect, our work was only done in that part of the stream very close to the outlet. Investigation work now being done at the University of Illinois is taking into account total air flow into the space and data concerning the counter-flow effect should result from this work.

Answering Mr. Zieber, the direct comparison of theory against practice was shown in the slide presented, but was not included in the paper. In other words, we calculated theoretical values after we found from test the average jet angle. A small amount of work was done on wide angle grilles, but we did not think it sufficient to add it to this paper. We are not in a position to make a definite statement at the present time

[1]Berechnung turbulenter Ausbreitungsvorgänge, by W. Tollmein, Zeitschrift für augewandte Mathematik und Mechanik, Band 6, Heft 6, Dec. 1926, pp. 468-478.

[2]Bericht über Untersuchungen zur ausgebildeten Turbulenz, by L. Prandtl, Zeitschrift für augewandte Mathematik und Mechanik, Band 5, 1925, pp. 136-139.

[3]Investigations of the Turbulent Mixing Regions Formed by Jets, by A. M. Kuethe. (*Transactions ASME*, Vol. 57, 1935, p. A-87).

[4]Some Two-Dimensional Aspects of the Ejector Problem, by J. A. Goff and C. H. Coogan. Paper presented at Dec. 1941 *ASME* Meeting—preprinted—to be published in Journal of App. Mech.

[5]Uber turbulente Strahlausbreitung, by E. Förthmann, Ingenieur-Archiv., vol. 5, no. 1, 1934, p. 42-54.

[6]Fundamentals of Hydro- and Aero-Mechanics, by Prandtl and Tietjens. (McGraw-Hill Book Co., Inc., New York, N. Y., 1934, pp. 233-240.)

regarding velocity and noise, except that we can state that the type or design of grille has a definite effect on noise.

We have not tested more than one slot at the face area so that we do not know the effect of the spacing of two narrow slots or of using two slots as an outlet.

Professor Coogan raised a question concerning pulsations which were observed. We did observe pulsations, but it was interesting to note that this work was carried on over quite a long period of time, and various men took readings. Some men would repeat the work that some of the others had done; but the average entrainment for each outlet, whether obtained by one man or another, agreed, although the reading of the velocity or the velometer reading in each square was not always the same; that is, one man would not get a reading in a certain square that was exactly the same as another man would get running the test at a different time. But the average throughout the whole traverse of the grid was very close.

With Vice-President Eastwood presiding the next paper entitled, Determining Sound Attenuation in Air Conditioning Systems, by D. A. Wilbur and R. F. Simons, was presented by Mr. Simons (complete paper will be published in May 1942 ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*). C. M. Ashley and H. K. Kunen, New York, N. Y., presented oral discussions and these will be published with the paper.

In the author's absence, F. N. Speller, Pittsburgh, Pa., presented the paper, Corrosion Tests in a Water-Recirculating Air Conditioning System, by W. Z. Friend (complete paper and discussion published in March 1942 ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*). A written discussion by C. M. Sterne, Long Island City, N. Y., was read, and Mr. Speller replied.

The final paper on the afternoon technical session, entitled The Performance of Stack Heads Equipped with Grilles, by D. W. Nelson, D. H. Lamb and G. E. Smedberg, was presented by A. F. Tuthill (complete paper published in January 1942 ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*).

SIXTH SESSION—

Wednesday, January 28, 9:30 A.M.

The first paper was presented by Dean Goff entitled, Vapor Pressure of Ice from 32 to —280 F (complete paper in February 1942 ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*).

L. P. HARRISON, Washington, D. C. (WRITTEN): The fundamental character of the thermodynamic relationships upon which Dean Goff has based his expression for the vapor pressure of ice, and the great accuracy of the basic data employed in the evaluation of the constants involved, leave little or no room for doubt as to the reliability of the vapor pressure values he has obtained within the degree of precision determined by the data in question.

The final test of the accuracy of the computed vapor pressure values is comparison of these data with carefully determined observed values or smoothed results of the vapor pressure observations. As is well known, the smoothed results are ordinarily derived with the aid of empirical equations or of equations having a partial thermodynamical ground where the constants are largely established by a least-square solution on the basis of observed vapor pressures.

The purpose of this note is to present some comparisons between Dean Goff's computed values and certain observed or smoothed values, in addition to Dr. Washburn's computed values. The results are shown in the appended table (complete table to be published in ASHVE TRANSACTIONS 1942), whose columns are explained in the succeeding descriptive remarks.

The agreement between the results of Dean Goff and of S. Weber is most remarkable, while the worst agreement is between the results of Dean Goff and those of Thiessen. The experimental determinations of Weber are considered to be more reliable than the values found by Thiessen from an empirical equation based on the observed vapor pressure data of Scheel and Heuse (Ann. d. Physik, vol. 29, 1909, p. 723).

Considerable confidence in Dean Goff's results is inspired by the comparisons presented, at least within the range of temperatures covered. This is exceedingly gratifying to all concerned.

PRESIDENT FLEISHER: I think this paper brings great credit to the Society, being one of the fundamental scientific developments in the thermodynamics of the art in which we are all engaged. I myself have felt for some time since Dean Goff agreed to carry on this work, and to credit it to our Society, that we certainly had acquired a distinction that this Society had not enjoyed before. In a way, Dean Goff's work compares with the fundamental work revolving around the development of the International Critical Tables carried on by the *American Society of Mechanical Engineers*. I really think it is the beginning of a movement to put us on a plane to which we have always aspired, but that we had not quite reached before. We owe a deep debt of gratitude to Dean Goff.

The next paper, Physiological Fatigue from Exposure to Hot Environments, by M. B. Ferderber, M.D., F. C. Houghten and W. L. Fleisher, was then presented by Dr. Ferderber (complete paper published in February 1942 ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*).

PRESIDENT FLEISHER: My interest in investigating the results reported in this paper led me to question the effectiveness of prognosticating the rise in leucocyte count as given in a lot of the data obtained at the Laboratory where I noted a lack of rise in leucocyte count in some of the high temperature work.

Some of the data which Dr. Ferderber has obtained indicated that those people who did not have a rise in leucocyte count corresponding to an increase in body temperature were not able to withstand a good many conditions imposed on them, such as the resultant recovery from various diseases.

From similar observations made at the Society Laboratory it was found that those subjects that did not have a rise in leucocyte count were not able to withstand the test and had to be withdrawn. I feel very definitely that Dr. Ferderber is not satisfied completely with his analysis or summary, but feels it is something that should be carefully investigated.

W. R. RHOTON: I like this paper because it is both theoretical and experimental, and as such the results can be applied in actual practice. Some time ago I was concerned with reference to high temperature problems and the effect on workers. In this particular application the temperatures were so high around the furnaces it scorched the clothing and in some cases even the hair. A method was devised to use a few panels consisting of copper tubing and metal lath with plaster on each side and water circulated through the tubes. These panels were installed in a rather inexpensive manner and we were able to reduce the temperature around the furnace to the extent that even a chilling effect was noted. We understood that the use of these panels even assisted in the length of life for the fire-brick located within the furnace.

L. T. AVERY: I do not know whether you realize the criticism the engineer must take when he undertakes to apply air conditioning to the health of the people. I recall an incident: a physician first called me to say, "Could we provide a humidity condition to take care of asthma?" and then said, "Well, we have decided to send the patient to Arizona, because we do not know what temperature or humidity we would like you to carry." Dr.

Ferderber has contributed to our Society, and we should be appreciative of this work because, while he may be a quasi-engineer, he is an air conditioning doctor.

The next paper entitled, Physiological Influence of Atmospheric Humidity, a report by the Research Technical Advisory Committee on Physiological Reactions was presented by Dr. C.-E. A. Winslow, the chairman (complete paper published in December 1941 ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*).

PRESIDENT FLEISHER (WRITTEN): This paper revives once more the moot question of the necessity for water vapor in the atmosphere, particularly in winter. One would gather from the paper that from the standpoint of thermal equilibrium the relative humidity of the air, in winter particularly, plays a very unimportant part. I feel that we are placing too much emphasis on thermal equilibrium as measured by rather crude devices or instruments. The sensation of dryness is distinctly unpleasant and in most instances calls for an increase in sensible temperature, which accentuates, due to the lower relative humidity, the uncomfortable feeling that accompanies too dry an atmosphere.

I have a very strong feeling that a certain amount of equilibrium is established possibly by electrical emanations between the body and the surrounding surfaces, and where the surfaces themselves are fairly dry and the surrounding atmosphere is dry, or we may say below a certain absolute humidity, the body is giving off certain fundamental electrical emanations which, under higher relative humidities, would not be given off. Just as we find in industrial applications that static electricity is eliminated at a fairly high relative humidity—say 50 per cent or over—so the body, possibly from the extremities (the fingers or other uncovered parts of the body, particularly sharp points), is disturbing essential sensory equilibria which would not occur at relative humidities above the points I have mentioned.

In a discussion of the effect of relative humidity on the feeling of comfort or the reactions of human beings to their environment with Professor Missenard in Paris several years ago, he claimed that our assumptions in connection with the effect of relative humidity had never been properly interpreted because we had never gone to the particular relative humidities which indicated change in reactions. He talked to me of 65 per cent relative humidity, which is a condition almost impossible to obtain in the wintertime in our cold climates, due to condensation on walls and windows. I am bringing this matter up simply because I do not believe that this subject can be dismissed without carrying the investigation to the optimum point, even though that particular point might not be practical.

From the angle of discussion of the Seeley paper,[7] I feel so definitely that this paper was an original contribution of tremendous importance, that it cannot be dismissed in a paragraph and that other features of this paper must be introduced into any discussion of this kind. In a discussion which I have had with Professor Seeley, we felt that the work that had to be done by the mucous membranes of the nose and throat was a function of the change in total heat between the introduced air and the resultant enthalpy of the air taken into the lungs. In other words, air saturated at 98 deg and 30 per cent relative humidity would have an enthalpy of 21.95 Btu per pound. Therefore, for each pound of air introduced, 45.97 Btu would have to be provided by the very small area of mucous membrane in the nasal passages. This might mean 20 per cent of the total thermal equilibrium that had to be made up by the whole surface of the body. Consequently, the relationship of percentages as set up by this paper must be oriented to the percentage of surface which has to accomplish the desired results. This particular amount of work, oriented against the areas involved, I think has not been properly appreciated. I therefore agree that this requirement for equilibrium is as much a function of cold, dry air as it is of hot, dry air.

There is one other point that I think has to be brought out in connection with this paper, a matter that I have discussed since presentation of a paper by me in 1931 on the effect of the pH concentration on the retention of moisture in a colloidal material. In this paper I brought out the fact that there were two kinds of moisture in any colloidal material, one being moisture of composition and the second being moisture of addition. The moisture of composition has an entirely different character than the moisture of addition. The moisture of addition is easily dissipated under normal temperatures. The moisture of composition, which is held tenaciously on the colloidal surfaces, can only be removed or dissipated under much greater bombardment, either electrical or thermal. In dealing with subjects under high dry-bulb temperatures, where the thermal equilibrium is entirely maintained by evaporation, a point is eventually reached where the moisture of addition is entirely or almost entirely dissipated. At this point it cannot be expected that the body can evaporate any moisture to maintain its thermal equilibrium, and this is probably the reason why men subjected to very high temperatures will function for a certain length of time without undue stress and then will succumb to coma.

It is therefore my contention that in all tests made at high temperatures and low humidities, unless there is definite regulation of the amount of liquid imbibed by the subject during the test and exactly the same amounts allotted to each subject, no definite conclusions can be drawn.

I suggest as a corollary to this report that this whole subject of the colloidal nature of the human body and its change in isoelectric state be determined by the constant observation of the moisture of addition factor as a determining point in the reaction of human beings to unfavorable environments.

G. DONALD FIFE, Washington, D. C. (WRITTEN): It seems that one of the conclusions to be drawn from this paper is that the physiological desirability of winter humidification is open to question. The statement that "There is, therefore, no physiological basis for humidifying air at moderate or low temperatures on thermal grounds" is not surprising, in view of the researches relative to effective temperatures. However, together with the section which treats of the influence of humidity upon the dryness of the nose and throat membranes, a case is presented for eliminating the use of humidifying equipment during the winter season. The humidities which can be maintained in the winter time are limited by the necessity of avoiding excessive condensation on the windows, and this paper shows that such limited humidities fall far short of the requirements necessary to avoid drying of the nose and throat membranes. The evidence offered is consistent with the statements in the chapter on Physiological Principles in THE GUIDE. On the other hand, the paper presented by Professor Seeley at the June, 1940 meeting suggests that humidification is beneficial to health as it reduces the sudden alteration in moisture loss from the nasal cavity when one goes indoors on a cold winter's day.

I believe that this question of the advantages of winter time humidification is very important and that the uncertainties arising from conflicting statements should be cleared up if possible. The prevalent belief, and one that has been sold to the public for a long time, is that indoor air should be humidified in the winter time; not only because it benefits the inanimate objects in the conditioned spaces—such as furniture and books—but because it is also beneficial to the health of the occupants. If it is considered that there is sufficient evidence available to refute this belief, it should receive the proper publicity. Perhaps there is a lot of steam for preheaters and power for spray pumps being wasted at the present time. Or, on the other hand, if the evidence is not sufficiently conclusive, it would seem that further investigations would be valuable.

C. J. STERMER, Chicago, Ill. (WRITTEN): This paper prompts a question as to whether some of the respiratory troubles attrib-

[7]Study of Changes in the Temperature and Water Vapor Content of Respired Air in the Nasal Cavity, by Lauren E. Seeley, (ASHVE TRANSACTIONS, Vol. 46, 1940.)

uted to highly heated dry rooms are due to the indoor condition, or the outdoor condition during the winter season.

Data here presented indicate rather conclusively that the absolute humidity might be the important factor to investigate from a health standpoint, rather than relative humidity.

In this connection it might be helpful to study the respiratory ailments if any occurring in cold storage plant workers, who spend considerable time during the summer season in a low atmospheric water vapor pressure environment and then immediately change to a high water vapor pressure and temperature environment when passing from the cold storage space to the outdoor summer air, as compared to a winter condition when these same individuals, as well as the general public in the northerly latitudes, live continuously in an environment of low water vapor pressure though changing from low to high temperature conditions.

DR. R. R. SAYERS, Washington, D. C. (WRITTEN): In connection with the data presented in this report it may be of interest to recall some of the results obtained by Bureau of Mines investigations in hot mines, especially with regard to the effects of exposure to hot, still, saturated air and hot, saturated moving air. Persons remaining at rest in saturated, still air at 91½ F for one hour experienced an increase in body temperature; a moderate increase in pulse rate; profuse sweating; and after effects of dizziness and weakness. In the same air conditions with air movement they experienced slight or no increase in body temperature; slight increase in pulse rate; slight perspiration; no after effects; and no ill effects at any time. Persons remaining at rest in saturated, still air at 95 F for one hour experienced an increase in body temperature; a marked increase in pulse rate; very profuse sweating, clothing being saturated with perspiration and sweat in shoes of all subjects; dizziness on movement, and increase in depth and rate of respiration (puffing somewhat on slight movement); chilly sensations in some subjects. Exposure to the same conditions with air movement caused slight or no rise in body temperature or pulse rate; profuse sweating, but not enough to wet all clothing; no untoward symptoms other than profuse sweating. Persons remaining at rest in saturated, still air at 100 F experienced a marked rise in body temperature, which reached 102.3 F; a marked rise in pulse rate, varying in different subjects from 152 to more than 175; profuse sweating, the shoes being partly filled with perspiration; and early appearance of dizziness, weakness, and persistence of symptoms for about one hour after test. The test was very trying. With air movement all the above symptoms were experienced and no subject remained a full hour in this atmosphere.

It is interesting to note that in one of our studies it was found that length of exposure to different high temperatures is a determining factor in the severity of the after depression produced in a subject. Although the physiological reactions may deviate further from the normal in higher temperatures for a short period of time than in a temperature not so high for a longer period, the fatigue resulting from the latter test is more prolonged than in the former. In an experiment in which the subjects remained for one and a half hours in the test chamber and the rectal temperature rose to 101.4 F and the pulse rate to 149 pulsations per minute, the fatigue and weakness which followed lasted for a longer period than the fatigue and weakness which followed in another experiment in which the rectal temperature rose to 103 F and the pulse to 162 with exposure for a period of only 45 min.

With regard to the effect of the dry and humid air on the mucous membranes, although the addition of humidity to the air in winter in air-conditioned buildings prevents this condition it apparently has no appreciable effect on the feeling of irritation in the throat which causes coughing in those suffering with colds. The drying effect of the warm air in the unconditioned, heated buildings is noticeable to those who return home to such an atmosphere after spending the day in an air-conditioned building where moisture has been added to the atmosphere. Probably most people have observed the drying effect on the mucous membranes of cold, dry air, especially with strenuous exercise in such air. With reference to the clinical practice of treating the membranes of persons suffering from bronchial diseases by soothing inhalations, it has been reported lately in some instances that medicine prescribed for coughs incident to colds has a considerable drying action on the mucous membranes of the mouth and throat.

DR. WINSLOW: I am glad that Mr. Fife called attention to the importance of this problem. If our observations as to the critical point of 0.4 inches of mercury are correct, it means that effective humidification in cold climates cannot be obtained. For the present I do not believe the results should be publicized, for I think the data should be checked by other investigators. The data on which these conclusions are based are presented in the last issue of the *American Journal of Hygiene*.

I was greatly interested in Mr. Fleisher's contribution. It is a little outside my field, but I would gladly accept him as an authority on that subject, because if anybody gave off electric vibrations from his fingers and every other part of his person, it is Walter Fleisher.

President Fleisher resumed the chair and expressed pleasure in introducing J. H. Walker, Second Vice-President of the Society, who was requested to preside. Mr. Walker stated that it was a real pleasure to be able to attend the Meeting and see everyone again. He then introduced W. F. Wells, who presented his paper, Sanitary Ventilation (complete paper published in the February 1942 ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*).

DR. HOLLAENDER: The author has done a great deal of pioneer work on the control of air-borne contagion. However, I feel that it would be helpful if a definition of bacteriocidal radiation would be stated in cooperation with the Council on Physical Therapy of the *American Medical Association*, who have just published an acceptance of the use of ultraviolet lamps for disinfecting purposes. It is extremely difficult to define the lethe, or accept it as Professor Wells has defined it, especially since it is not simple to have standard suspension of an organism. This also applies to the other definitions which Professor Wells has given.

PROFESSOR WELLS: I think Dr. Hollaender's insistence on complementing the report of the Council on Physical Therapy is quite proper. The Council has recently defined the strength of burners which are acceptable to the *AMA*, but disclaims the responsibility, if I understand their report, for defining dosage which determines the breathing quality of confined atmospheres. My paper attempts a system of units by which dosage can be defined—a first attempt, but one necessary before reporting upon clinical results of radiant disinfection.

It is much more simple to determine the dose of a drug by the amount in a pill or a spoonful, and it is not difficult to compute or to test the concentration of a chemical disinfectant in the atmosphere. It is more difficult to specify the density of a barrage, though this also is necessary if you are to prevent penetration. However, the physics of ultraviolet emission can be described and the geometry of irradiation between a source and a barrier which absorbs the lethal photons can also be accurately specified. Lethal uniformity of radiation in space cannot be defined with physical precision, nor can the individual paths of drifting bacteria be predicted. A statistical average can however be specified just as pressure specifies the statistical average of molecular impingements. The paper attempts to formulate the results of these statistical integrations.

It is also quite true that standard suspension of standard organisms is not a physically or chemically perfect procedure. Standardization of disinfectants has long been a subject of argument among bacteriologists, due to the nature of biological processes and phenomena. However, it was necessary to stand-

ardize disinfectants for intelligent application, and it will be equally necessary to standardize radiant disinfection. Hair-splitting details have not prevented progress in air-conditioning, nor should they impede progress in sanitary ventilation.

The results of radiant disinfection in the schools are evident to every mother of the school children—to the school teachers, the school boards, and others in the communities who have witnessed the experiments. It is not necessary that they go into statistical details on immunity or susceptibility, or await the pronouncements of a super-board of experts. It was obvious that during the largest epidemic of measles in Philadelphia the unirradiated upper grades suffered far more heavily than the irradiated lower grades. In previous years in these schools measles has attacked the primary school age (characteristic of measles since the dawn of medicine)—five or six in these ages for one in High School ages. For the first time the age incidence of measles has been reversed.

This experience convinces us that ventilating engineers stand at the threshold of a new phase of their art. Just as it was necessary for water supply engineers to absorb sanitary specifications into their design, so has it become necessary for ventilating engineers to accept hygienic specifications. Engineers must play a part equal in importance to that assumed by the medical profession in the pronouncement of the Council on Physical Therapy.

SEVENTH SESSION—

Wednesday, January 28, 2:00 P.M.

The joint session with the *National Warm Air Heating and Air Conditioning Association* was opened by President Fleisher, who introduced C. A. Olsen, retiring president of the Association. After a few remarks by Mr. Olsen, George Boeddener, Managing Director of the Association, was introduced, as well as Allen W. Williams, the former Managing Director, who is also a Life Member of the Society.

R. R. Tucker, Smoke Commissioner for the City of St. Louis, gave an address on Smoke Control. He traced the history of the problems in the elimination of smoke from the City, which finally culminated in 1937 with the passage of a new ordinance which included a clause demanding that all fuel below 2 in. in size be washed until the ash content did not exceed 12 per cent. He stated this provision in itself did not attempt to reduce the smoke problem, but it was the beginning of the present program because it gave some measure of control over the type of fuel to be used. Subsequently, the ordinance was revised providing that all fuels should contain less than 23 per cent volatile matter on a dry basis, and if one desired to burn a high volatile coal, it was necessary to use some form of mechanical burning equipment. Mr. Tucker said that largely due to the cooperation of the citizens and many varied interests of a community, the St. Louis smoke ordinance has been quite successful.

QUESTION: How do you handle the automatic firing of oil burners which are improperly adjusted?

MR. TUCKER: Our smoke regulations cover both oil burners and stokers.

MR. RHOTON: In the passage of this ordinance is it not true that people have actually saved money instead of putting them to additional expense?

MR. TUCKER: This is a controversial question. A survey conducted in St. Louis in 1929 indicated that the smoke problem cost the citizens about $19,000,000.00, which would mean that each individual was paying from $20.00 to $25.00 a year to live in the polluted atmosphere of smoke, which are admittedly hidden costs. In dealing with the general public it means little for me to say that we have reduced the soot deposit from 400 tons per square mile to 300, but if the soot is not on the window sills or on the porches, then the homeowner begins to believe that definite improvements have resulted.

The question of costs has been a very vital factor, primarily due to the extreme variation in the heat content of the fuel. Coals which are now being brought into St. Louis will vary from 13,000 to 15,000 Btu per pound, which means that there is 30 to 50 per cent more heat per pound of coal as compared to the fuel previously burned, which had heat contents as low as 9,000 Btu per pound.

L. D. PARSONS, JR., Washington, D. C.: I would like to inquire if you specify or control in any way the boiler or burner settings?

MR. TUCKER: Yes, this is done and in the case of power installations each one is considered a separate and distinct application.

R. A. SHERMAN, Columbus, Ohio: Basically Mr. Tucker is working on two principles for the elimination of smoke, one being the use of smokeless fuels and the other the use of equipment which will not produce smoke. Incidentally, the St. Louis smoke campaign has helped the stoker industry a great deal, but for the individuals in the lower income bracket who are forced to purchase fuels brought in from considerable distance, I am not sure whether these individuals feel that the increased cost has been justified.

In reality the effect of the St. Louis ordinance has been to awaken the bituminous coal industry to the fact that something must be done to utilize the available coals that lie within 10 miles of St. Louis, but which cannot be burned in the present automatic equipment. At the present time the coal industry has a program under way which is progressing to the point where some of this fuel can be burned smokelessly.

Vice-President Eastwood assumed the chair and introduced C. F. Warner who presented the paper entitled, Pressure Loss Caused by Elbows in 8-in. Round Ventilating Duct, by M. C. Stuart, Mr. Warner and W. C. Roberts (complete paper published in October 1941 ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*).

QUESTION: What is the minimum radius of elbow which is safe without having the pockets?

MR. WARNER: We have found that a 2-in. inside radius is about the smallest we can get for an 8-in. circular duct.

A. P. KRATZ, Urbana, Ill.: I think most of our experimental work is indicated on the basis of inside radius which has much more bearing on the pressure loss through an elbow than the outside radius. Some of your slides seem to indicate exactly the reverse, which I would appreciate having you reconcile.

MR. WARNER: The pictures were taken with square 7-in. ducts, and round on the outside. Previous investigations show about a 20 per cent reduction in pressure loss by having the outside square but we could not obtain this result in our test. We used one which was square on the outside, with a 2-in. inside radius that had approximately the same loss as that with the radius on the outside.

QUESTION: In the last picture shown has the effect of splitters been investigated?

MR. WARNER: This phase of the investigation has not been completed, but we are now installing curved vanes in the model. The pressure reduction in square ducts with vanes seems to be quite large as compared to round ducts. We have some data which indicated that we can reduce the loss as much as 65 per cent with the proper introduction of turning vanes in a square duct.

The next paper entitled, Observed Performance of Some Experimental Chimneys, by R. S. Dill, Paul R. Achenbach and Jesse T. Duck, was presented by Mr. Dill (complete paper and discussion published in this issue, see p. 252). Several oral comments were of-

fered by H. F. Randolph, Utica, N. Y., R. K. Thulman, Washington, D. C., C. E. Olsen, Elyria, Ohio, and Allen J. Johnson, Primos, Pa.

The final paper entitled, Heat Loss Through Basement Walls and Floors, by F. C. Houghten, S. I. Taimuty, Carl Gutberlet and C. J. Brown (complete paper published in January 1942 ASHVE JOURNAL SECTION, *Heating, Piping & Air Conditioning*), was presented by Dr. Houghten.

P. D. CLOSE, Chicago, Ill. (WRITTEN): As has been apparent for some time, and is evident from this paper, the allowances to be made for basement heat losses should be reduced considerably from present values. While there may not as yet be sufficient data to make detailed recommendations, it seems that certain approximations are justified from the results of the paper, especially for the locality and conditions of the tests. The actual floor heat loss was less than 10 per cent of the calculated value and for the conditions of the test, the *total* heat loss averaged less than 2.0 Btu per square foot per hour through the uninsulated floor. I believe this would be a reasonable allowance to make for an average uninsulated floor for a heated basement used for living quarters or recreational purposes. For a temperature difference of say 20 deg between the air and the ground, the over-all floor coefficient even without insulation would be only 0.10. Apparently it would not be necessary to make any floor allowance for an unheated basement.

The heat loss through basement walls and floors apparently follows an entirely different pattern than is the case with walls exposed to the outside weather where steady state heat transfer is involved. With basement floors and walls there is a process of diffusion of heat into the earth and the extent of the diffusion increases as the heating season progresses until a maximum depth is reached. All of the dirt under the floor extending to the maximum limit of the heat diffusion, may be considered as contributing to the heat resistance of the floor. It appears that the depth of the diffusion of heat from the basement will increase as the conductivity of the earth increases and this increased depth of diffusion would, in turn, have the effect of offsetting the increase in conductivity of the earth.

Since the rate of heat flow through the wall varies with the depth below the ground level, the wall heat loss allowance should, of course, vary with the depth, although an average figure would probably be sufficiently accurate in most cases. The wall heat loss varied from 1.9 to 4.8 Btu per hour per square foot on the uninsulated side and from 1.7 to 3.9 per hour per square foot on the insulated side, but it should be remembered that the top of the simulated basement was covered with 18 in. of dirt so that the wall depth was greater than it would normally be. The effect of the insulation in this case is less pronounced than it would be in the case of an exposed wall because of the insulating value of the considerable thickness or depth of dirt. It would seem, however, from these tests that the *average* allowance for the wall heat loss (below grade) for a basement heated to 70 F need not be more than about 3.0 Btu per hour per square foot or a *maximum* of about 5.0 Btu per hour per square foot at the top of the grade level.

E. C. LLOYD, Lancaster, Pa. (WRITTEN): The results confirm the practice that no heat loss need be figured for floors over unheated basement spaces. Also they warrant a reduction in the heat loss figured through floors of basements used for home activities.

Admitting that these investigations have covered only a small simulated basement and that somewhat different results might come with varying soil conditions and changing moisture conditions in the soil due to seasonable changes, drainage and water levels more or less constant in location, it would seem that the results secured thus far could be applied to varying soil and moisture conditions if conductivities were available for representative soil samples in wet and dry condition. It would seem that the procurement of necessary soil samples and the determination of a K value for them dry and with varying water content, or possibly simply dry and saturated with water, should not represent an undue amount of work to supplement the excellent start already made.

The increase in conductivity of soils by reason of moisture has been brought out by earlier experimentation, but it is unfortunate that a better description of the materials previously tested is not available. A summary of soil conductivities other than those mentioned in the paper may be given as follows:

Soil, dry — K = 0.958	Lees & Chorlton—*ASRE*-Ins. Comm. Rep. 1922
Soil, wet — K = 4.64	Lees & Chorlton—*ASRE*-Ins. Comm. Rep. 1922
Soil, normal condition (excavated in Munich) including stones 1 to 3 in.	Density 127.4 lb/cubic feet K (68 F) 3.63 Groeber
River Sand, fine grained, dried completely by heat	Density 94.8 lb/cubic feet K (68 F) 2.26 Groeber
Same, with normal moisture content (6.9 per cent by weight, 11 per cent by volume)	Density 102.4 lb/cubic feet K (68 F) 7.825 Groeber

That the range in results is considerable is evident, but it may develop on further test that the differences except for those brought about by varying moisture content are inconsequent.

The importance of further consideration of condensation on the inside of these walls is a point that may well be emphasized. The increasing use of basement space for game rooms, rumpus rooms, home work shops, etc., as well as the trend toward air conditioning of such spaces brings condensation troubles in basements to the front. Some study of the effect of the addition of insulating finishes to the interior wall might well be considered at this time.

C. H. PESTERFIELD, East Lansing, Mich. (WRITTEN): I feel that we are indebted to the Research Laboratory and the authors for the work that has been done on this subject, but I feel, as probably Dr. Houghten does, that this paper is not the final solution to this problem. It is doubtful if any precise solution can ever be made due to the magnitude of variables existing in work of this nature.

It is unfortunate that the test room was not made somewhat greater in wall dimensions, for my experience on this subject has shown, for a heated basement, that at a distance of 8 ft from the outside walls that little, if any, heat is actually lost through the floor. Do not curves for position D in Fig. 8 in this work pretty fairly indicate the same conclusion?

Is it not a fact, Dr. Houghten, that it would be impossible to place the thermocouples in an isothermal plane due to the fact that the actual isothermal is inclined and not horizontal? Further, that the greater the distance from the wall in either direction, the plane changes angle.

It is gratifying to note that the data in this paper substantiate the belief that many of our engineers have had for years, that the heat loss through the basement has been grossly exaggerated; however, few expected that the actual loss and the calculated loss differed so greatly.

E. F. DAWSON, Norman, Okla. (WRITTEN): This paper presents information that is a very important addition to our present building heat loss data and places on a scientific basis the determining of design heat losses through basement walls and floors. These heat loss calculations up to the present have, in many cases, been made on assumptions based chiefly on guess work or incomplete test data.

The test conditions were, in general, quite representative. The climatic conditions at the test location, Pittsburgh, Pennsylvania, is representative of many sections of the country. The test room built for this research project, with its side walls entirely below grade is typical of many basement rooms and below grade rooms of large buildings. The use of both insulated and uninsulated walls and floors in the construction of this test room furnishes valuable data for comparing heat flow, temperature and condensation effects.

The graphical results shown in Fig. 5 present interesting and valuable information relative to the soil temperatures at depths from 9 to 72 in. for a 12 month period. These observations made at a station 21 ft from the structure wall were not noticeably influenced by the heated basement room, and hence might serve

as a guide in estimating soil temperatures for intermittent heating design. Also, by comparing similar data at other locations with these soil temperatures we will be aided in our heat loss estimates at these locations.

The information portrayed on Figs. 6, 7 and 8 giving the logs of the air temperature 6 in. from the inside surfaces and the soil temperatures 44 in. from these surfaces will be invaluable in studying air to soil temperatures and arriving at representative temperature differentials for heat loss determinations through basement walls and floors.

The soil temperature gradient around the perimeter of the structure together with the heat flow at 7 observation points is shown in Fig. 9 for a typical heating design day. The soil temperature values in the vertical stations of this perimeter may be useful in estimating soil temperatures of other basement rooms whose walls extend to a greater or less depth below grade level.

The discrepancy between the *rule of thumb* method, which gave 13,730 Btu per hour as the heat loss for this room and the determined heat loss by test of 1750 Btu per hour is such as to indicate the great need for this more scientific method of approach.

The variables that will need to be considered for the general application of basement room losses are: soil characteristics, soil temperature gradients in warmer and colder localities and basement rooms that have part of their walls above grade level.

The elimination of condensation appears to be, as stated by the authors, a matter of applying additional heat resisting materials to the walls and floors.

A. B. ALGREN, Minneapolis, Minn. (WRITTEN): I would like to commend the authors on the contribution they have made in starting to clarify the question as to ground temperatures. Although the results obtained would possibly apply to many localities, they also indicate that further studies are necessary and should, if possible, be made with respect to different geographical locations. A ground temperature of 50 F, a value that we assume for heat loss calculations, appears to be about correct for our locality. This again is based more upon assumption than actual test values.

I do feel that this work should be extended to other localities, not only for its value in accurately calculating basement losses, but mainly with respect to the condensation problem in basements. With definite data on ground or earth temperatures, the mechanics required for eliminating condensation would then become quite simple.

DR. HOUGHTEN: The only question that requires an answer was one by Professor Pesterfield, regarding the isothermal plane. It is true that the isothermal plane surrounding the basement and down in the earth is not horizontal. About 2 ft of the thermocouple in the bottom of the tube was coiled up so as to be in a horizontal position, but actually this is not strictly the isothermal plane, although it is not very far from it.

Installation of Officers

W. T. Jones, Boston, Mass., past president of the Society, conducted the installation of officers and those inducted into office were E. O. Eastwood, Seattle, Wash., *President;* M. F. Blankin, Philadelphia, Pa., *First Vice-President;* S. H. Downs, Kalamazoo, Mich., *Second Vice-President;* and E. K. Campbell, Kansas City, Mo., *Treasurer.* The newly elected Council members were L. G. Miller, East Lansing, Mich., A. J. Offner, New York, N. Y., A. E. Stacey, Jr., New York, N. Y., and B. M. Woods, Berkeley, Calif.

L. T. Avery, Chairman of the Resolutions Committee, offered the following resolutions:

Resolutions

WHEREAS, the 48th Annual Meeting of the ASHVE is to be concluded tomorrow without any further business session and

WHEREAS, the Technical Sessions, the Entertainment, Committee Meetings, and business transacted have been outstanding in interest and stimulation to those attending,

BE IT RESOLVED THAT an expression of thanks and appreciation be adopted and copies thereof be transmitted to each of the persons and agencies who have contributed to make this meeting so enjoyable;

To the National Society Officers and Committees and Past President Fleisher, who devoted so much of their time and energy to the work of the Society during the past year;

To the Officers and Members of Philadelphia Chapter and their ladies who have been so cordial in their welcome and so hospitable to everyone;

To Merrill F. Blankin, General Chairman and to Mrs. Blankin, Chairman of the Ladies Committee, who have been so considerate of our every wish;

To the Chairmen and Members of the several committees who have given so generously of their time to assure the successful carrying through of the Program;

To the authors and their associates who prepared and presented the fine technical papers, each of which is an outstanding contribution to the art and science of handling air;

To the Management and Staff of the Bellevue-Stratford Hotel;

To the Philadelphia Convention and Tourist Bureau for their cooperation;

To the Newspapers of Philadelphia for their fine coverage of this meeting and to the Trade Magazines who have cooperated so ably and generously;

To the Hon. Bernard Samuel, Mayor of Philadelphia and his official delegate, Mr. Charles Grakelow, for his inspiring speech of welcome to the City of Philadelphia;

To Edgar A. Kirby, District Chief of the London Fire Force, for his able speech and the interesting pictures given at our Monday luncheon.

To Paul Fleming and his associates for a marvelously mysterious Evening of Magic;

To Mr. Roth and his Associates who labored so arduously for the 7th International Heating and Ventilating Exposition which was postponed because of the war declaration;

To Past President Jones for his able installation of our new Officers; and BE IT FURTHER RESOLVED THAT in anticipation of an enjoyable banquet tonight we express our hearty appreciation to Stanley High and those who will make this an inspiring and a notable event; and

To all others who have contributed in so many ways to our enjoyment of the 48th Annual Meeting.

Respectfully submitted,
RESOLUTIONS COMMITTEE,
L. T. Avery, *Chairman,*
C. E. Bentley,
Brig.-Gen. W. A. Danielson.

At 4:45 p. m. on Wednesday, January 28, President Eastwood adjourned the 48th Annual Meeting of the Society.

NATIONAL WARM AIR CONVENTION

The 28th annual convention of the *National Warm Air Heating and Air Conditioning Association* was held in Philadelphia at the Benjamin Franklin Hotel, January 27-28, 1942, and the committee activities and discussions reflected the influence of the war on the association's affairs. Many of the technical reports dealt with problems resulting from requests for assistance from government departments.

W. W. Timmis, chief of the plumbing and heating branch of the War Production Board, explained how the Civilian Supply Section was being reorganized by setting up industry branches with each branch taking care of its own industry. He urged heating engineers to familiarize themselves with Process Priority Orders PD-1 and the new PD 25a, and advised that a special division of the branch is to be set up to secure information on process orders. In connection with the simplification program, the WPB is asking the cooperation of industry in compiling information which will be used as a basis for the promulga-

tion of a Simplification Order which will sharply limit the number and sizes of many items. He mentioned the shortage of copper, and expressed his opinion that the year 1942 is the critical year and every effort must and will be made to conserve resources to the utmost.

Prof. A. P. Kratz, University of Illinois, talked on the new Gravity Heating Manual and stressed the importance of figuring all heat losses accurately to insure adequacy in furnace capacity. S. Konzo, also from the University of Illinois, addressed the meeting on his recent activities in connection with government defense housing and army heating.

In his official report to the Corps of Engineers, Professor Konzo recommended a more careful check of installations prior to acceptance of the heating plants.

The following officers were elected for 1942: *President*—H. S. Sharp, Cleveland, O.; *First Vice-President*—H. P. Mueller, Milwaukee, Wis.; *Second Vice-President*—F. E. Mehrings, Peoria, Ill.; *Managing Director* and *Treasurer*—George Boeddener. *Directors*: C. Ackerson, Gadsden, Ala.; R. W. Blanchard, Holland, Mich.; E. B. Lau, Dayton, O.; P. S. Miller, Middletown, O.; M. D. Rose, Pittsburgh, Pa.; A. L. Rybolt, Ashland, O.; F. G. Sedgwick, Minneapolis, Minn.; and Atlee Wise, Akron, O.

COOLING TOWER INDUSTRY AND COMBUSTION TOPICS AT KANSAS CITY

February 2, 1942. Pres. Gustav Nottberg opened the February meeting of the Kansas City Chapter, held at the Hotel President, which was attended by 28 members and guests. The minutes of the previous meeting were read and approved.

It was decided to change the regular meeting date of the Chapter from the first to the second Monday of each month.

W. A. Russell reported on the 48th Annual Meeting of the Society held in Philadelphia, which was also attended by Messrs. E. K. Campbell and M. M. Rivard, representing the Chapter.

President Nottberg then introduced L. T. Mart, Kansas City, Kans., who presented a complete and interesting picture, with the aid of slides, of the growth of the cooling tower industry. His discussion clearly pointed out that the trends in the industry were toward mechanical induced draft towers. He presented figures indicating how, in the face of increased costs of material and labor, the actual costs of cooling tower equipment had been reduced because of modern design. After his talk, Mr. Mart answered several questions in a general discussion, followed by adjournment.

January 5, 1942. The Kansas City Chapter held its monthly meeting at the President Hotel, with 25 members and guests attending. Following the reading of the minutes of the previous meeting, which were accepted as read, a report by the treasurer was given.

Mr. Campbell then suggested that a full year's program be arranged as soon as practical.

Harry Atwater was introduced as the speaker of the evening. His talk covering the aspects of combustion of gas, oil, and coal in different types of heating boilers was enthusiastically received by the members. An interesting discussion ensued with answers to various questions ably given by Mr. Atwater.

NORTH CAROLINA HOLDS WINTER MEETING

February 27, 1942. The North Carolina Chapter held its winter dinner meeting at Duke University, Durham, N. C., with 19 members and guests in attendance.

The speaker of the evening, T. C. Cooke, heating and air conditioning engineer, spoke on radiant heating. Mr. Cooke recently designed and supervised the installation of a panel heating system in a new home in Durham, and he discussed the problems encountered in the design of the system, its installation and its operation. An interesting discussion by the members followed and it was considered that much valuable information was exchanged.

During the business portion of the meeting a nominating committee was elected consisting of Messrs. E. R. Harding, Harry Hoffman, Arvin Page, F. J. Reed and W. M. Wallace, II.

It was announced that the annual Chapter meeting would be held in Charlotte, N. C., with H. H. Hill in charge of arrangements.

WESTERN MICHIGAN OIL AND NATURAL GAS SITUATION

February 9, 1942. The February meeting of the Western Michigan Chapter was held at the Rowe Hotel, Grand Rapids, and was called to order by Pres. W. G. Schlichting.

Following a report by the treasurer, Prof. L. G. Miller and T. D. Stafford gave an interesting account of the 48th Annual Meeting of the Society in Philadelphia. Prof. C. H. Pesterfield then read a paper on the effect of attic fan operation on the cooling of a structure, and also one on heat loss through basement walls and floors, both of which were presented at the Annual Meeting. J. W. Miller gave a report on the paper on the performance of some experimental chimneys, while B. F. McLouth reported on the paper performance of stack heads equipped with grilles; he also discussed the paper on the pressure loss caused by elbows in 8-in. round ventilating ducts.

After some discussion on a paper dealing with the physiological influence of environmental conditions by Professor Miller, it was suggested that some thought be given to just what kind of a program the members would prefer for the field day to be held some time in May.

January 12, 1942. The January meeting was held at the Rowe Hotel, and in spite of bad weather conditions there were 28 members and guests present. President Schlichting called the meeting to order and in the absence of the secretary the reading of the minutes was dispensed with.

President Schlichting then announced the appointment of Mr. Stafford as Chapter Delegate and Mr. McLouth as representative on the Nominating Committee.

A motion was made and carried that Charles H. Morton be sent a paid-up dues card together with letter of transmittal. Another motion was made and carried that any member going into service be sent a paid-up Chapter dues card for each year that he is in the service, together with a letter of transmittal by the secretary.

James Spindle was then intro-

duced, and those present obtained a great deal of first-hand information as to the present conditions in the oil and natural gas situation in Michigan. Mr. Spindle has a world of information on this subject and together with his map and chart gave a very splendid discourse, interesting and entertaining, as well as educational. Mr. McKinstry, one of his associates, then showed how he conducted an orsat test and why it was so necessary to do this with all gas-fired heating jobs. This was most interesting, especially to those in localities where natural gas is just beginning to be supplied.

SMOKE ORDINANCE DISCUSSED AT PITTSBURGH

February 9, 1942. The meeting was called to order in the Hotel Roosevelt, with 26 members and guests in attendance. Following the reading of the minutes and the treasurer's report, Prof. C. M. Humphreys, chairman of the program committee, reported that the subject for the March meeting would be corrosion in heating and air conditioning systems.

E. C. Smyers, as chapter delegate to the 48th Annual Meeting, gave a report of the Chapter Delegates' meeting held in Philadelphia.

J. F. Collins, Jr., reported on the work of the Council of the Society and mentioned a proposed method for heat loss calculations.

President Smyers then appointed A. F. Metzger as chairman of the auditing committee, and R. H. Sweeney to assist him.

Professor Humphreys introduced Prof. S. B. Ely, formerly associate professor of mechanical engineering at Carnegie Institute of Technology and now superintendent of Bureau of Smoke Prevention of the City of Pittsburgh. Professor Ely spoke on the work of his Bureau and on the operation of Pittsburgh's new smoke ordinance. He pointed out that in spite of the relatively low cost of natural gas in this district it would still be necessary for many people to use the even less expensive coal. He also brought out that previous campaigns to educate people in the smokeless combustion of bituminous coals had been unsuccessful, hence, it seems that the solution of the local smoke problem will require the use of more stokers, the burning of coke or processed low volatile fuels, or the use of magazine type of furnaces. Extensive discussion followed, before adjournment.

PRESIDENT EASTWOOD VISITS ILLINOIS

February 2, 1942. National President's night was observed by the Illinois Chapter at their February meeting held at the La Salle Hotel, Chicago, with 70 members and guests in attendance.

Pres. I. E. Brooke called the meeting to order and introduced two past presidents of the Society, H. M. Hart and John Howatt, and then presented Prof. E. O. Eastwood, Seattle, Wash., newly installed president of the Society.

Professor Eastwood gave a brief message to the Chapter from the new Society Officers, asking the Chapter's cooperation through the coming year and then went into the subject of the evening, Heating, Ventilation and Air Conditioning of Modern Aircraft. President Eastwood showed slides tracing the development of the modern airplane from those used in 1918 to the modern flying fortress.

He further explained the different methods used in controlling temperature and humidity conditions in modern aircraft, including information on high altitude flying requiring supercharging of the interior of the plane in order to support life. He then discussed the design and operation of various wind tunnels throughout the United States used for testing and design of planes and plane models, and showed slides of these tunnels together with the methods used in obtaining and recording data.

Professor Eastwood gave a clear and authoritative picture of the subject and an interesting discussion followed.

January 12, 1942. The January meeting was called to order by President Brooke with 52 members and guests present. The December minutes were read and approved, followed by a brief talk by M. W. Bishop about the 48th Annual Meeting program.

Lester Seelig introduced the speaker of the evening, Walter C. Keys, mechanical products engineer, who spoke on the subject of rubber mounting for ventilating equipment. Mr. Keys pointed out that rubber mountings have definite advantages and definite limitations when applied to heating and ventilating equipment. His talk was of a practical nature and was well received and enjoyed by the Illinois Chapter members. He included simple demonstrations of principles, as the absorption of vibration, which he demonstrated by supporting a weight on a string of several rubber bands looped together, the upper end being held in the hand. By selecting a weight whose natural frequency is quite low, and by moving the supporting band up and down at a rate several times faster than the natural vibration of the weight, it was shown by Mr. Keys that the latter would remain almost still, although being supported by the rapidly moving hand. During the usual discussion period a number of questions were answered by Mr. Keys.

The meeting adjourned at 9:45 p.m. with a rising vote of thanks to Mr. Keys for his interesting and instructive talk.

EFFECTS OF WAR HEARD AT ONTARIO

February 2, 1942. The February meeting of the Ontario Chapter was held at the Royal York Hotel, Toronto, with 47 members and guests attending.

Prior to the meeting the members and guests met at the plant of the Standard Sanitary and Dominion Radiator Co., Ltd., where they inspected the pottery and foundry operations.

Pres. C. Tasker called the meeting to order, and following the introduction of seven new members, he reported on the recent 48th Annual Meeting of the Society in Philadelphia, which he attended. He outlined the activities of the new Research House on hot water and steam heating, and also reported on the discussion on the nominating committee.

G. C. Crawford, president of the Standard Sanitary and Dominion Radiator Co., Ltd., was then introduced. He gave a brief talk on the effect of the War on foundry products. Following Mr. Crawford's address there was a short discussion, and S. W. Alexander extended a vote of thanks to Mr. Crawford on behalf of the members and guests for his delightful address.

IOWA MEETINGS

January 13, 1942. Pres. Perry La Rue presided at the January meeting of the Iowa Chapter, which was held at the Kirkwood Hotel, Des Moines, with 33 members and guests in attendance.

Following a short business meeting, J. M. Appleton, resident engineer of one of the ordnance plants, gave a very interesting talk on the problems confronting the designers of this plant, particularly with regard to the heating of the large spaces and the distribution of steam in overhead pipe lines. Interesting highlights were pointed out by Mr. Appleton. He mentioned that of the number of people this plant will employ, about half of them will be in what is known as the hazardous explosive area. One of the difficulties, he pointed out, was that they were required to omit copper in this area and use steel, which presented many complicated problems.

One interesting point disclosed by Mr. Appleton was in the method of temporary heating in which corn dryers were used and by-products of combustion were discharged into the buildings; the large areas and plenty of ventilation nullified any bad effects.

An interesting discussion took place, followed by adjournment.

December 16, 1941. The December meeting was held at the Russell Lamson Hotel, Waterloo, with President La Rue presiding and an attendance of 24 members and guests. The minutes of the previous meeting were read and approved.

F. K. Knapp and Richard Gehr were introduced, and gave a very interesting talk on panel heating, especially as connected with the use of wrought-iron pipe. Slides were shown illustrating many jobs showing the use of wrought-iron pipe and the methods of fabrication. Several questions were asked during the discussion which followed, and were answered by the speakers.

Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 42 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by April 15, 1942, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ALLEN, WILLIAM A., Vice-Pres., Sprague & Sprague, Inc., Pittsburgh, Pa. (*Reinstatement*)	F. E. Lynn L. W. Cordrey (*Non-member*)	G. O. Weddell E. C. Smyers
BACON, WILLIAM H., JR., Mech. Engr., Gresham Realty Co., New York, N. Y.	J. E. Muir (*Non-member*) J. A. Lambertine (*ASME*)	Dr. Edward M. Cannon (*Non-member*) A. T. Kniffen (*Non-member*)
BRODIE, AARON H., Pres., J. Brodie & Son, Inc., Detroit, Mich.	F. X. Marzolf Fred Kaiser	F. C. Purcell W. H. Old
CHAPPELL, HENRY D., Htg. & Vtg. Engr., Burroughs Adding Machine Co., Detroit, Mich. (*Reinstatement*)	G. H. Cummins J. S. Kilner	H. E. Ziel N. B. Hubbard
CLARK, JAMES R., Asst. Chief Engr., Page & Co., Charlotte, N. C.	H. H. Hill C. E. Petty	R. M. Warren, Jr. M. F. Wooten, Jr.
COE, SEYMOUR A., Student, Pratt Institute, Brooklyn, N. Y.	A. A. Collard (*Non-member*) A. W. Doll (*Non-member*)	H. S. Cameron (*Non-member*) R. B. Dale (*Non-member*)
DANOWITZ, CHESTER J., Asst. Naval Archt., Supervisor of Shipbuilding, U.S.N., Camden, N. J.	Lt. Comdr. David Ladd N. A. White (*Non-member*)	Lt. Comdr. Charles Tooke (*Non-member*) Paul Greetin (*Non-member*)
DEFLON, JAMES G., Cooling Tower Engr., The Fluor Corp., Los Angeles, Calif.	Leo Hungerford H. H. Douglas	J. B. Griffith H. H. Bullock
DOMINY, CHARLES B., JR. Engr., Puget Sound Navy Yard, Bremerton, Wash.	F. J. Pratt R. J. Davis	I. A. Naman L. L. Bysom
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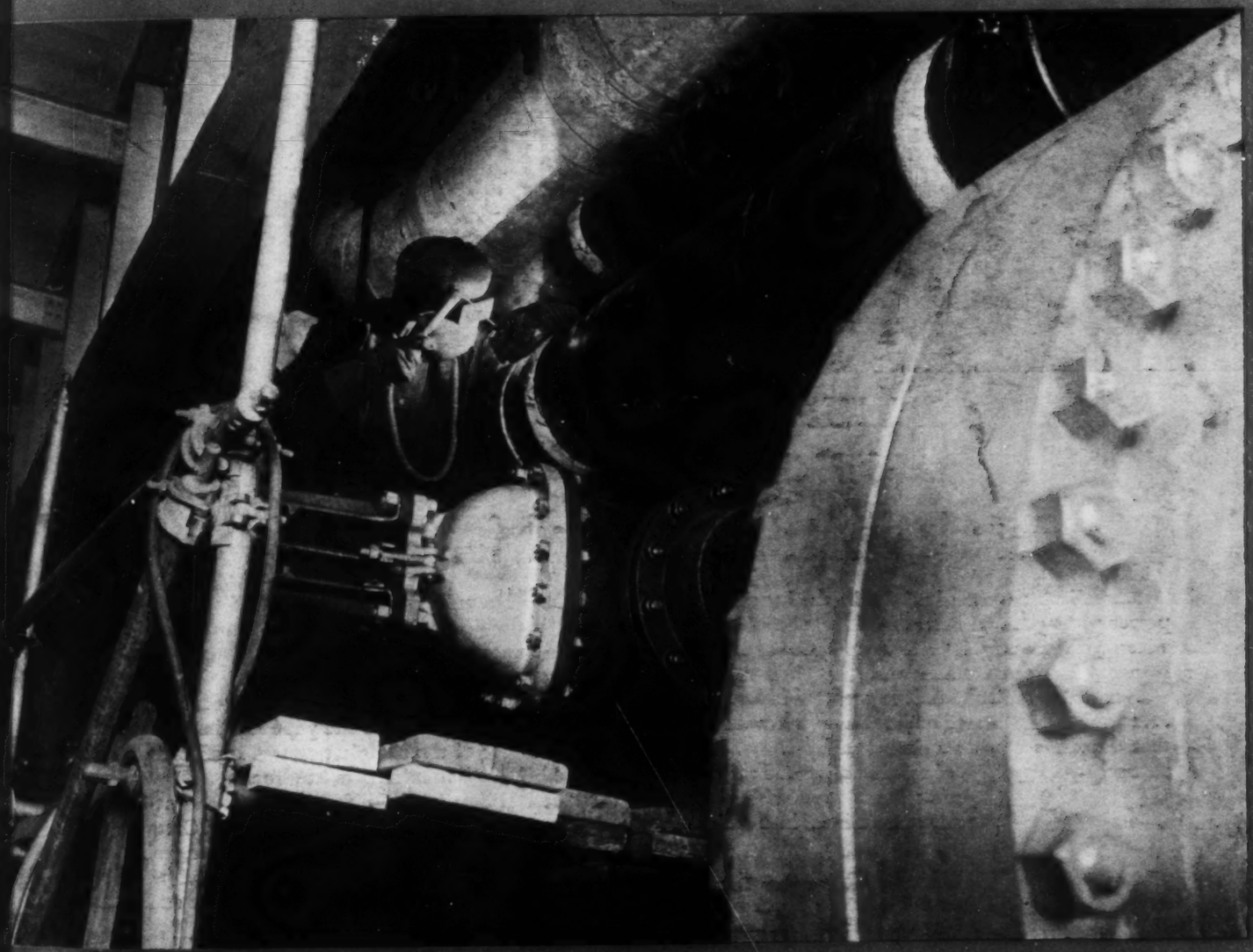

The section on fabrication details of the new code for pressure piping is described this month

CONTENTS..PAGE 43